Energy audit of Captive Power Plants at ACC Ltd, Keymore

Prepared for The Associated Cement Companies Ltd, Keymore (MP)

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We greatly acknowledge the support of entire engineering and maintenance staff of old and new CPP in completing the field measurements during energy audit.

Executive summary

The main report contains detailed analysis and recommendations emerging from the study of the plant. A summary of the major recommendations has been given in the following sections.

Boilers and Turbines

- New CPP is equipped with three atmospheric fluidised bed combustion boilers (AFBC) supplied by ABB-ABL Ltd. Boilers are designed to generate 66 TPH of superheated steam at 66 Kg/cm^2 pressure and 485°C at 100% MCR.

  The efficiency of boiler #1 and #2 is worked out to be 81.2 and 82.45% respectively.

  Air infiltration was observed in boiler#1between boiler and air heater, which is measured to be 22% of the stoichiometric air requirement.

  It is recommended to prevent all such infiltration which could be from flanges, joints, open inspection windows, etc. This will increase the flue gas temperature to the level of no air infiltration (i.e. 154°C) from 135°C.

  It has been calculated that there would be an annual fuel saving of around 392 MT of coal. The corresponding monetary saving would be Rs 4.7 lakhs with almost nil investment towards plugging the infiltration.
- Old CPP is equipped with four boilers. Out of the four boilers, one FBC and one stoker boiler were in operation during the study.

  The efficiency of boiler #2, 3 and 4 is worked out to be 75.31, 68.45 and 68.70% respectively.
- Excess air in all the old CPP boilers was on the higher side, this is because of more opening of FD air and air infiltration in the system.

  It is beneficial to maintain the desired level of excess air in the boiler which will result in the improvement of boiler's efficiency levels and fetch recurring monitoring savings.
- It has been calculated that there would be an annual saving of Rs 20.51, 10.77 and 9.82 lakhs in boiler #2, 3 and 4 respectively through plugging air leakages and heat recovery in economiser or in additional heat recovery unit if the flue gas temperature does not fall below 200°C.
- Instrumentation is poor for old CPP boilers. Steam flow and coal meters installed for FBC boilers do not function properly.

It is recommended to rectify all the available control instruments on the panel for FBC and stocker fired boilers.

- For new CPP, the unit heat rate obtained on the basis of turbine heat rate and accounting for boiler and generator efficiency works out to 3277 kCal/kWh. This compares well with the average heat rate of 3159 obtained after taking into account the effect of dumped steam.
- Operating values of all the turbines are well within the accepted norms.

Pumps and fans

- A study was conducted to assess the performance of the pumps under the prevailing operating conditions.
  The efficiency of boiler feed and condensate extraction pump is worked out to be 61.2 and 70.5% respectively.
- Techno-economics of installing variable speed AC drive instead of conventional throttling in condensate extraction pump is worked out. The estimated annual savings are to the tune of 49,600 kWh (i.e. Rs 79,400/- per year) with simple payback period of 6.8 years. As long term option, this proposal can be taken up.
- The efficiency of ID and FD fans of stoker fired boilers is worked out to be 60 and 45% respectively.
- Presently outlet dampers are used for controlling the air flow of fans in old CPP. It is recommended to replace these with variable speed AC drive system to achieve an efficient and stepless flow regulation. An estimated saving of Rs 85,000 can be achieved with an investment of Rs 4,65,000 yielding a simple payback period of 5 years for one FD fan.
  Discussion with plant personnel revealed that they are planning to convert the stoker-fired boilers to FBC in near future. In that case, it is recommended that use of variable speed drive for FD/ID fans should not be considered at present because the economics as well as configuration of the drive may change after conversion to FBC boiler.

Compressed air systems

- There are nine reciprocating type air compressors operating in the captive power plant (CPP). Free air delivery test was performed on four reciprocating compressors (No. #2, #3, #4 in NCPP and No. #1 connected with vessel/ receiver 2 in OCPP). The free air discharge of compressor 2 and 4 is found to be below the satisfactory level. It is, therefore, recommended to get compressor #2 and #4 in NCPP overhauled to have better productivity and lower energy consumption.

- The leakage test, to quantify the leakages in the distribution lines of ash handling system 2 and 3 of NCPP was carried out. The leakage rate in the compressed air line 2 and 3 is calculated to be 9.5% and 6.7% respectively. By plugging these leakages, there would be an annual electrical energy saving of 2430 kWh and 940 kWh. The corresponding monetary saving would be Rs 4900 and Rs 1900 respectively.
- Temperature of the air supplied to the suction of compressor #1 in OCPP should be monitored periodically to ensure that it is, at least, equal to the ambient temperature. In case any rise is observed, the inlet air duct can be extended further to outside of room.
- It was observed that five (3 number in NCPP and 2 number in OCPP) air receivers drain valve was manually operated to facilitate the water drainage. This provision not only passes the water but also allow the compressed air to pass to the atmosphere intermittently.
  It is recommended to use automatic drain valve to reduce the wastage of compressed air.

Cooling towers

- The captive power plant of the cement manufacturing complex is equipped with two large sized cooling towers. For optimum performance, range should be as high as possible and approach should be as low as possible. The electricity savings have been estimated with respect to the following measures:
  By replacing the existing fan blades with hollow FRP blades in the cooling tower #1 and #2, a saving of 29330 kWh and 28670 kWh or Rs. 58,700 per annum and Rs. 57,400 per annum is possible with a payback period of around 9 to 15 months.
  It is recommended to install the CT-ENSAVER+ system (thermostatic controller) in the cooling tower #1 and #2 fitted with old CPP and new CPP respectively. The savings in the cooling tower #1 and #2 because of this will be 59996 kWh and 58706 kWh or Rs 1.19 lacs per annum and Rs.1.17 lacs per annum.

Coal handling system

- The fines in the crushed coal, below 3 m size, are far more in quantity than the value for which boilers have been designed.
- It is observed that present feed rate is 50 tons/hr as against belt capacity of 140 ton/hr. CHP has to be run for two shifts. If the feed rate improve to 80-100 tons, coal feeding can be accomplished in one shift.

Water chemistry

- DM plant comprises of two steams of 8 m^3 and 9.5 m^3 per hr capacities. The sequence is cataion, degaser, anion, and mixed bed.
- It is recommended to have intermittent blow down instead of continuous blow down.
- Dosing of anti-corrosion chemical should be reviewed by assessing the efficacy of the chemical added.

Energy analysis

Various issues reported by the plant have been analysed based on the data collected during the energy audit as well as the log book data. These are as follows:

- The auxiliary power consumption is high as the power plant is operated at load factor of 60 to 80%. It will come down to about 14.5% at 100% load. Moreover, (i) the power distribution loss of 0.37% is also included in this, (ii) two boilers are operated even at 50% load. However, power consumption by cooling tower and coal handling plant can be reduced through the measures suggested. It can be further reduced by about 1% if operation with dump steam is avoided.
- The main reasons for the high unburnt combustibles in the fly ash is (i) high percentage of fines in the coal fed to the boilers, (ii) overbed coal feeding system, and (iii) small free-board height of the furnace. It is practically not possible to control fines in the coal. But modification for underbed feeding and increasing free-board will have to be carried out if combustibles in the fly ash are to be brought down to 4 to 6% range.
- The plant heat rate can be improved only if the power plant is operated without dumping steam. For this, proper load management of the cement plant (particularly the raw mill) should be taken up on highest priority.
- For further improvement of power factor beyond existing value of 0.88, a detailed system study with respect to dynamics of all connected loads

should be carried out. However, the present value of 0.88 is considered good for a well managed system.

- Operating experience should be gained through practice for operating the boilers with 50% of the bed in dumped condition. For this, it should be ensured that inlet air dampers are air-tight in the closed condition, for the dumped bed section.
- For integrated operation of new and old CPPs, an additional valve in the steam line and an additional line in the condensate recycle circuit will have to be incorporated as shown in Figure 8.7.1.2.
- For improving the operation management of the plant and reduce number of start-ups and shut-downs, it is necessary to develop 'Operational and Performance Monitoring Guidelines' taking into account the capability of the power plant and the dynamics of existing loads. Suggestions for this are given in sections 8.8 and 8.9.

1

Introduction

Comprehensive energy audit of Captive Power Plants of Keymore Cement Works (KCW) was carried out by Tata Energy Research Institute (TERI), to study the present energy consumption pattern and to identify the energy saving possibilities in various sections of the plant. TERI team visited the plant in October 2001 to undertake measurements and for collection of data. Discussions were also held with the plant personnel to accomplish the task in the most meaningful manner.

The KCW has cement production capacity of 2.2 MTPA comprising of two plants of 1 and 1.2 MTPA. The average requirement of the KCW is 38 MW. To meet the power requirement, KCW has its own Captive Power Plant (CPP) comprising of old and new plant. The old and new plant can be interconnected steam to steam and on power to power side. Both the plants are not connected to grid and therefore, operate in radial mode.

The focus of the present study was to carry out a detailed analysis of the following major energy consuming sections.

- Boilers and turbines
- Pumps and fans
- Compressed air system
- Cooling towers
- Coal handling system
- Water chemistry
- Energy analysis

A draft report covering each of these areas is given in chapters 2 to 8.

Besides these main equipments, the Plant Manager brought out following problems in the plant:

- High heat rate
- High auxiliary power consumption
- High percentage of combustible in fly ash

All the three issues have been addressed in Chapter 8 on 'Energy Analysis', which presents analysis, findings and recommendations for minimising the energy consumption.

2

Boilers and turbines

2.1 New CPP boilers

New CPP of KCW is equipped with three atmospheric fluidised bed combustion boilers (AFBC) supplied by ABB-ABL Ltd. Boilers are designed to generate 66 TPH of superheated steam at 66 Kg/cm^2 pressure and 485°C at 100% MCR. Two boilers operate at a time and cater to a 25 MW turbine to generate power. The main consumers of power are cement mills (pulverisers). Large variations are observed in the power demand of these cement mills and due to this, steam demand is also fluctuating in a large range (50%-100% of MCR) depending on the load on these mills. Whenever the steam demand reduces due to reduction in load because of tripping/maintenance of the mill, it is passed to feed water tank through condenser after de-superheating. Steam dumping valve remains open most of the time, however, percentage opening keep changing. The unit has to be operated in this condition to be in a state of readiness for the quick load pick-up when mill is switched on.

Boiler#3 has an interconnection with old CPP and supplies steam to run one of the turbo-generator there. Steam from the boiler#3 is supplied to old CPP after pressure reduction and de-superheating.

All the three boilers of new CPP are identical. Each boiler is equipped with two superheaters with attemperator in between, economiser and air pre-heater. There are 4 coal feeders in each boiler, arranged in the front wall which supply the fuel to furnace bed. There is provision of three HSD fired burners in one of the sidewall for the boiler light up. Each boiler is equipped with two FD fans for fluidisation and combustion in bed; two secondary air fans for combustion support; two cooling air fans for cooling of peep holes, feeders, etc; one burner air fan to supply combustion air to oil burners during the boiler start-ups. To provide balance draft in furnace by taking suction from ESP outlet, there are two ID fans in each boiler.

Past data shows that the coal consumption in new CPP boilers varies in a range of 8.8-9.4 TPH whereas the steam to coal ratio varies between 5-5.2. The lowest and highest steam to coal ratios are observed in boiler#3 which are 4.5 in July and 5.7 in the month of May.

2.1.1 Fuel

The main fuel is coal. HSD is used for boiler light up only. Boilers are designed for over feed coal firing and fuel is spreaded over the bed with the help of stoker. Coal and ash samples were collected during the study and the analysis is given below. Table 2.1.1 shows the coal analysis and the unburnt in the bottom ash and fly ash.

Table 2.1.1 Details of coal analysis

Elements	*Value (%)*
Moisture	8.5
Ash	32.8
Carbon	47
Hydrogen	4
Sulfur	0.55
Nitrogen	0.22
Oxygen	6.8
GCV(kCal/kg)	4680
Average coal sizing	100% through 10 mm
(Sieve analysis)	>65% through 3.3 mm
	>35% through 1mm
Details of ash analysis	
Unburnt in bottom ash	4.5%
Unburnt in fly ash	14.4%

2.1.2 Feed water system

After expansion in turbine, the exhaust steam goes to condenser and from here condensate is collected in hot well at about 45°C. From here, it goes to de-aerator through LP heater. The water temperature raises to about 80-100°C. In deaerator (mechanical type), water gets heated up to 135-145°C. From deaerator, feedwater goes to high-pressure heater for which the steam is also extracted from the turbine. At last, the feed water enters the economiser at about 170-190°C and finally reaches to the boiler drum at 225-235°C. 5-10% variations are also observed in these temperature data which are linked with the variations in the turbine loading. Condensate return is almost 100% and the boiler make up water is very less (<3%).

LP steam and HP steam are extracted from the turbine at 0.8-1.3 kg/cm^2 (a) (at 90-105°C) and at about 11-14 kg/cm^2(a) (at 280-300°C) respectively.

2.1.3 *Blow down*

Both continuous and intermittent blow down is practised in new CPP. Continuous blow down valves in each boiler are kept crack open and intermittent blowdown is performed on the basis of boiler water analysis provided by the plant laboratory. All blowdown lines are connected to a closed vessel for the respective boilers. Flash steam is not utilised and therefore vented off. The blowdown TDS were reported to be less then 50 ppm which are within the prescribed range.

2.1.4 *Boiler efficiency*

The boiler efficiency can be computed either by direct method from the heat input and output or by the losses (indirect) method. The reliability and accuracy of the fuel flow and steam flow measurements are very critical for the direct method, which will be quite misleading some times. Hence, indirect method is chosen to compute the actual efficiency. ASME method is used for efficiency calculation.

The heat losses from the boilers include

- Dry gas loss which is a function of excess air and the flue gas composition and the corresponding gas temperature at air heater outlet. Corrections has been applied to AH outlet flue gas temperatures (for boiler#1) considering air ingress in the system. This loss was established by taking flue gas analysis after the boiler, economiser and after air heater.
- Losses due to combustion of hydrogen and moisture, which depends on % H_2 and moisture in fuel; moisture in air, etc; excess air and exit gas temperature.
- Unburnt carbon losses which depends on the unburnt in bottom and flyash, and the weight ratios of these ashes.
- Sensible heat loss in bottom and fly ash
- Radiation losses

2.1.5 Basic input data for computing boiler efficiency

The important parameters measured/collected from the plant for the performance evaluation of the boilers by indirect method, are given below:

1. Oxygen percentage in flue gas
2. Carbon monoxide level in flue gas
3. Flue gas temperature
4. Drum pressure
5. Feed water temperature
6. Steam flow rate
7. Temperature of boiler surfaces
8. Analysis of fuel and ash
9. Ambient conditions (temperature & relative humidity)

2.1.6 Observations and recommendations

The flue gas composition & temperature, ambient conditions and other relevant parameters for evaluating the efficiency of the boilers were measured using portable instruments at site. Wherever it is not possible to take measurements by using the portable instruments, panel readings are used for the calculations. For each boiler (boiler#1 and boiler#2), flue gas readings were taken at three places (after boiler, economiser and airheater) to ensure the consistency. It was also tried to keep the boiler load constant to about 50-55TPH (75 to 83% MCR) during the flue gas measurement. Extensive surveys of the surface temperatures in different boiler sections were also conducted for computing the structural losses and ascertaining the condition of insulation. The summary of the performance of boiler#1 & #2, as evaluated by the indirect method, are given in table 2.1.6, the details are given at Annexure 2.1.

Table 2.1.6 Performance of new CPP boilers

Parameters	*Boiler #1*	*Boiler #2*
Measurements & observation		
Oxygen content in flue gas* (%)	3.25	4.0
Excess air level* (%)	17	24
CO level (ppm)@	250	450
Outlet flue gas temperature (°C)	154**	123
Ambient conditions		
Dry bulb temperature - 30°C		
Wet bulb temperature - 25.3°C		
Heat balance		
Dry flue gas loss (%)	4.94	3.89
Loss due to CO in flue gas (%)	negligible	negligible

Parameters	Boiler #1	Boiler #2
Loss due to moisture in air (%)	0.16	0.13
Loss due to hydrogen and moisture (%)⇓	6.10	5.97
Loss due to unburnt in bottom ash (%)	0.64	0.64
Loss due to unburnt in flyash (%)	6.15	6.15
Loss due to sensible heat in bottom ash (%)	0.25	0.25
Loss due to sensible heat in fly ash (%)	0.16	0.12
Structural loss (%)	0.39	0.39
Total heat losses (%)	18.8	17.55
Boiler efficiency (%)	81.2	82.45

* after boiler

@ before ID

** corrected for air ingress. Actual measured flue gas temperature was 135°C after the air heater for boiler 1. For boiler 2, no air ingress was noticed.

⇓ includes the moisture of combustion air, fuel and moisture due to H_2 content of the fuel

It is clear from the above table that biggest loss is unburnt in fly ash followed by the loss due to hydrogen and moisture in the fuel. These losses are discussed in the detail below.

2.1.6.1 Loss due to unburnt in fly ash

This is the highest loss in both the boilers. During the audit, it was observed that both the boilers are facing the problem of high unburnt in fly ash. To check the combustible levels in the ash, fly and bottom ash samples were collected and analysed in KCW laboratory. The combustible levels are reported to be 4.5% and 14.4% in bottom and fly ash respectively. However, the sieve analysis for the coal being fed to the boilers, carried out during the visit, gives the following result:

Table 2.1.6.1 Sieve analysis

Sieve size	+10mm	+8mm	+6.3mm	+3.3mm	+2mm	+1mm	-1mm
Boiler#1	Nil	3.3	9.3	24.6	17.9	12.4	32.5
Boiler#2	Nil	3.7	7.4	17	17.6	14.1	40.2
Average	Nil	3.5	8.4	20.8	17.8	13.3	36.4

A detailed analysis of the causes of high unburnt and the recommendations for improvement, is given in Section 8.3.

2.1.6.2 Loss due to hydrogen and moisture in fuel & atmosphere

This is the second biggest loss and depends upon the moisture and hydrogen content of the fuel and flue gas exit temperature. The moisture and hydrogen content of the fuel is fixed and, therefore, the only controlling factor is the exit flue gas temperature. The loss due to hydrogen and moisture is worked out to be 6.1% and 5.97% in boiler#1 and #2, respectively. This loss will marginally be reduced to a level of 6.02 % in boiler#1 after plugging the infiltration of air.

2.1.6.3 Dry flue gas loss

The dry flue gas loss mainly depends on two factors namely a) excess air level and b) flue gas temperature at the exit. These two factors are discussed in detail below.

2.1.6.3.1 Excess air level

Excess air is one of the important parameter in determination of boiler's performance. Every fuel needs a specified quantity of stoichiometric air for its combustion. In actual practice, since, mixing of fuel with air is never perfect, a certain amount of excess air is always needed to complete the combustion and ensure the release of entire heat contained in the fuel. Too much of air results in excessive heat loss as the surplus air does not take part in combustion but carries away heat to the atmosphere from the boiler furnace. Besides, surplus air lowers the furnace temperature and thereby the heat transfers rate, and hence results in lower combustion efficiency.

Likewise, if the excess air is too less than the optimum quantity, combustion would be incomplete resulting in the formation of gases like carbon monoxide etc, in the flue gas. The maximum permissible limit for CO is 0.1%. Above this, loss due to CO formation is much higher than the benefit availed on account of lower excess air level.

As clear from the table 2.1.6.1, excess air level in all the boilers is quite reasonable, in a range of 15-25%. In boiler#2, excess air level was slightly high compared to the boiler#1. This could be due to variation in load conditions of the boiler#2 during the measurement.

2.1.6.3.2 Flue gas temperature

Flue gas temperature is another major contributor to the dry flue gas loss and also has the impact on other losses. The flue gas temperatures are measured in the boiler#1 & #2 are 154°C and 123°C.The flue gas temperature in the boiler#1 is on higher side whereas the same in boiler#2 is OK. In fact, air infiltration was observed in boiler#1between boiler and air heater, which is measured to be 22% of the stoichiometric air requirement. Any air leakage in the flue gas system reduces the heating efficiency and increases the load on ID fan. Moreover, it also reduces the flue gas temperature which is measured to be 135°C after air heater. The flue gas temperature in boiler#1 is adjusted for this (in the absence of air infiltration) which is 154°C. It is recommended to prevent all such infiltration which could be from flanges, joints, open inspection windows, etc. This will increase the flue gas temperature to the level of no air infiltration (i.e. 154°C) from 135°C. It is expected that the difference in temperature (19°C) will be utilised for enhanced heat transfer in economiser or air heater and final outlet temperature would be 135°C. The increase in efficiency and coal savings due to this are shown in the table 2.1.6.3.2 below, and the details are given at Annexure 2.2.

Table 2.1.6.3.2 Improvement in efficiency

	Efficiency with air infiltration (%)	Efficiency without air infiltration (%)	Coal savings/ annum (MT)	*Monetary savings (lac Rs)*
Boiler#1	81.2	82.1	392	4.7

Savings are calculated on the basis of average monthly coal consumption data. This would be the recurring saving per annum with almost nil investment towards plugging the infiltration.

2.1.6.4 Loss due to unburnt in bottom ash

This loss is 0.64% only in both the boilers and it depends on the unburnt content of the bottom ash. Bottom ash samples were collected during the plant visit and sent to the KCW laboratory for determination of combustible, which is tested to be 4.5% for both the boilers.

2.1.6.5 Radiation loss

The structural losses in both the boilers is calculated to be 0.39% which is well within the limits. However, few spots like inspection doors and windows need special attention as the temperature is too high there.

2.1.6.6 Sensible heat loss in ash

This heat loss is due sensible heat carried away in bottom ash and flyash and depends on the bottom ash disposal and flue gas temperature. These losses are 0.2% and 0.12% respectively for bottom and fly ash.

2.1.6.7 *Loss due to Moisture in air*

This loss is because of moisture content in air, which depends on the relative humidity and the ambient temperature. The other controllable factors, on which this loss depends, are excess air level and flue gas temperature. Presently, this loss is calculated to be 0.16% and 0.13% in boiler#1 and #2, respectively and it will be reduced, in boiler#1, to 0.14 after plugging the air infiltration.

2.2 Old CPP boilers

Old captive power plant of KCW is equipped with four boilers. Two of them are atmospheric fluidised bed combustion boilers (converted from stoker to FBC) and other two stoker fired. All the boilers of old CPP are supplied by Steinmuller GmBH, Germany. Out of the four boilers, one FBC and one stoker fired, were in operation during the study. Steam generation from boilers vary in a large range depending upon the electrical load requirement of the cement mills. These boilers cater to two turbines of 6 MW each and one of the capacity 8.25 MW. Old CPP also has an interconnection from boiler#3 of new CPP and gets steam to run one of the turbo-generator. The details of these boilers are given in the table 2.2.

Table 2.2 Details of old power plant's boiler

Description	Boiler#1 & #2	Boiler#3 & #4
Type	AFBC	Stoker fired
Installed	1954	1954 & 1958
Design		
Pressure	31 Kg/cm^2	30 Kg/cm^2
Temperature	438±5°C	440°C
Steam flow	25 MT/hr	20 MT/hr
Actual		
Pressure	27±1 Kg/cm^2	27±1 Kg/cm^2
Temperature	400±10°C	400±10°C
Steam flow	8-25 MT/hr	4-10 MT/hr
FD fans	2 per boiler	2 per boiler
ID fans	1 per boiler	combined
Heat recovery unit	Economiser	Economiser

Large variation in steam demand is quite frequent (as evident from the table 2.2) which depends on the power requirement of the cement mills. Whenever the steam demand reduces, the excess steam, after desuperheating, sent to feed water tank through condenser.

2.2.1 FBC boilers

Both the FBC boilers are converted from the stocker firing. Each boiler is equipped with three banks of superheating tubes without attemperator in between, one economiser, two FD fan and one ID fan. The furnace in each boiler is divided in to two sections namely upper and lower plenum; one FD fan dedicated to each plenum. ID fan is common to both the plenums. Almost 80-85% of total air is supplied as primary air to maintain the bed height. Capacity control by controlling the FD is not practised in old CPP and it is controlled in steps i.e. 0%, 50% and 100% of the boiler capacity. There is no control between the steps as reduction in air quantity may disturb the bed, as stated by plant personnel. Whenever steam demand is reduced from these boilers they operate with high excess, air unless the load reduction is 50%.

Capacity test was performed on the boiler#2 for four hours with a view to establish the steam generation and fuel consumption as steam flow meters and coal consumption counters were not functioning properly during the study. The boiler#2 was operated at almost full load during the test. The steam generation and coal consumption were worked out to be 23.3 T/hr and 5.72T/hr respectively. Specific steam generation is calculated to be 4.07 T of steam per ton of coal.

2.2.2 Stoker fired boilers

Earlier all the four boilers were stoker fired but two have been converted to FBC boilers. These boilers have one superheater bank, two FD fan and common ID fan. Since both the boilers have the common ID fan, draught conditions in both the boilers have to be same as change in settings of one boiler has the bearing on the other. Whenever the FD fan setting of one of the boiler is changed, it has to be followed by change in the ID damper setting of the same boiler and the FD/ID fan settings of other boiler.

As per the performance test carried out on these boilers, steam generation is 7.3t/hr and 7.7 T/hr respectively for boiler#3 and boiler#4. Coal consumption was measured to be 1.9 T/hr and 1.75 T/hr for boiler#3 & #4 respectively.

2.2.3 Fuel

Coal is the fuel in the boilers. The FBC boilers are designed for over-bed coal firing and the fuel is spread over the bed with the help of stoker. Coal is fed to the stoker by screw feeders. During the study, mismatch in coal size was observed in the stoker fired boilers. The size of coal was varying from fines of 3mm to the big lumps of 10 cm, particularly in the stocker-fired boilers. Coal and ash samples were collected during the study and the analysis is given below. Table 2.2.3 shows the coal analysis.

Table 2.2.3 Details of coal analysis (as fired)

Elements	FBC boiler#2	Boiler#3 and #4
Moisture	9.3	8.5
Ash	32.3	33.4
Carbon	46.4	46.2
Hydrogen	4.1	3.9
Sulfur	0.63	0.64
Nitrogen	0.22	0.22
Oxygen	6.5	7.0
GCV(Kcal/kg)	4685	4538
Average coal sizing	>95% through 10 mm	3 mm - 10cm
(Sieve analysis)	>65% through 3.3 mm	
	35% through 1mm	
Unburnt in bottom ash	0.4%	14.9
Unburnt in fly ash	15.3%	-

2.2.4 *Feed water system*

After expansion in turbine, the exhaust steam goes to condenser. The condensate is collected in hot well at about 55±10°C. From here, it goes to economiser and finally reaches to the boiler drum at 155±10°C. Little variations are also observed in these temperature data which are also linked with the turbine loading. Condensate return is almost 100% and the boiler make up water is very less (3-4%).

2.2.5 *Blow down*

Intermittent blow down is practised in old CPP. Blow down is performed on the basis of boiler water analysis provided by the plant laboratory. All blow down lines are connected to a closed vessel for the respective boilers. The blow down TDS were reported to be varying in a range of 1000-2500 ppm.

2.2.6 *Boiler efficiency*

The boiler efficiency can be computed either by direct method from the heat input and output or by the losses (indirect) method. The reliability and accuracy of the fuel oil flow and steam flow measurements are very critical for the direct method, which will be quite misleading some times. Hence, indirect method is chosen to compute the actual efficiency. ASME method is used for efficiency calculation.

The heat losses from the boilers include

- Dry gas loss which is a function of excess air and the flue gas composition and the corresponding gas temperature at air heater outlet. Corrections has been applied to AH outlet flue gas temperatures (for all the three boilers) considering air ingress in the system. This loss was established by taking flue gas analysis after the boiler, economiser and after air heater.
- Losses due to combustion of hydrogen and moisture which depends on % H_2 and moisture in fuel; moisture in air, etc; excess air and exit gas temperature.
- Unburnt carbon losses which depends on the unburnt in bottom and flyash.
- Sensible heat loss in bottom and fly ash
- Radiation losses

.2.7 *Basic input data for computing boiler efficiency*

The important parameters measured/collected from the plant for the performance evaluation of the boilers by indirect method, are given below.

1. Oxygen percentage in flue gas
2. Carbon monoxide level in flue gas
3. Flue gas temperature
4. Drum pressure
5. Feed water temperature
6. Steam flow rate
7. Temperature of boiler surfaces
8. Analysis of fuel and ash
9. Ambient conditions (temperature & relative humidity)

.2.8 *Performance of boilers*

The flue gas composition & temperature, ambient conditions and other relevant parameters for evaluating the efficiency of the boilers were measured using portable instruments at site. Wherever it is not possible to take measurements by using the portable instruments, panel readings are used for the calculations. For each boiler flue gas readings were taken at two places (after boiler and economiser or before ID) to ensure the consistency. These measurements were taken during the boiler performance test when the boiler loads were almost constant. Extensive surveys of the surface temperatures in different boiler sections were also conducted for computing the structural losses and ascertaining the condition of insulation. The summary of the performance of boilers, as evaluated by the indirect method, are given in Table 2.2.8, the details are given in Annexure 2.3.

Table 2.2.8 Performance of OCPP boilers

Parameters	*Boiler #2*	*Boiler #3*	*Boiler #4*
Measurements & observation			
Oxygen content in flue gas* (%)	6.05	8.44	10.28
Excess air level*(%)	38	64	91
CO level (ppm)@	1837	38	38
Outlet flue gas temperature (°C)**	251	288	251
Ambient conditions			
Dry bulb temperature - 30°C			
Wet bulb temperature - 25.3°C			
Heat balance			
Dry flue gas loss (%)	10.87	15.22	15.10
Loss due to CO in flue gas (%)	0.01	negl.	negl.
Loss due to moisture in air (%)	0.35	0.49	0.49
Loss due to hydrogen and moisture (%)⇓	6.66	6.72	6.56
Loss due to unburnt in bottom ash (%)	0.06	8.86	8 86
Loss due to unburnt in flyash (%)	6.39	0	0
Loss due to sensible heat in bottom ash (%)	0.12	0.26	0.26
Loss due to sensible heat in fly ash (%)	0.22	0	0
Structural loss (%)	0.50	0.50	0.50
Total heat losses (%)	24.69	31.55	31.27
Boiler efficiency (%)	75.31	68.45	68.73

* after boiler

@ before ID

** corrected for air ingress. Actual measured flue gas temperatures were 202°C, 130°C and 130°C before the ID fan in boiler#2, #3 and #4 respectively.

⇓ includes the moisture of combustion air, fuel and moisture due to H_2 content of the fuel

2.2.9 *Observations and recommendations*

It is clear from the table that biggest loss is dry flue gas loss followed by loss due to unburnt in ash and the loss due to hydrogen and moisture in the fuel. These losses are discussed in the detail below.

2.2.9.1. Dry flue gas loss

The dry flue gas loss mainly depends on two factors namely a) excess air level and b) flue gas temperature at the exit. These two factors are discussed in detail below

2.2.9.1.1 Excess air level

Excess air in all the old CPP boilers was on the higher side, this is because of more opening of FD air and air infiltration in the system. The difference in excess air at boiler and before ID fan is given in Table 2.2.9.1.1 below.

Table 2.2.9.1.1 Air infiltration

	After boiler		Before ID	
	Oxygen (%)	*Excess air (%)*	*Oxygen (%)*	*Excess air (%)*
Boiler#2	6.05	38	9.73	82
Boiler#3	8.44	64	16.1	312
Boiler#4	10.28	91	16.1	312

It is clear from the above table that there is lot of air ingress in the system. Ingressed air, however, does not facilitate the combustion but dilute the flue gas and because of this overall flue gas temperature reduces. This reduction of flue gas temperature reduces the heat transfer in the economiser and the valuable heat is lost in the atmosphere.

Excess air in the FBC boiler is also on the higher side. Excess air for FBC boiler should be in a range of 15-30% where as the same for the stocker fired boiler should be in a range of 40-60%. It is beneficial to maintain the desired level of excess air in the boiler which will result in the improvement of boiler's efficiency levels and fetch recurring monitoring savings.

2.2.9.1.2 Flue gas temperature

Flue gas temperature is another major contributor to the dry flue gas loss and also has the impact on other losses. The measured flue gas temperatures before ID fan is 202°C for boiler#2 and 130°C for the boiler#3 and #4. Since there is heavy air leakage in the boiler system (table 2.2.9.1.1) it will not be correct to use these temperatures for efficiency calculations. Any air leakage (of cold air) in the flue gas system reduces the flue gas temperature and heating efficiency and increases the load on ID fan. Therefore, these temperatures are corrected for no cold air infiltration conditions. The measured and corrected temperatures are mentioned in the Table 2.2.9.1.2a below.

Table 2.2.9.1.2a Air infiltration

	Flue gas temperature before ID fan (°C)	
	Existing condition	*Case of no infiltration*
Boiler#2	202	251
Boiler#3	130	288
Boiler#4	130	251

There is a great difference in the actual measured temperatures with air ingress and same corrected for no air ingress condition. It is recommended to utilise this difference in economiser or in additional heat recovery units if the flue gas temperatures do not fall below 200°C. By a thumb rule, reduction by 20-22°C in flue gas temperature will improve the boiler efficiency by 1%. It is suggested to maintain flue gas temperature about 150°C which will be same as maintained in the new power plant.

During the study, air ingress were observed at many places. Few major points are listed below:

- openings between the tube banks of economiser in all the boilers
- through the ash pendulum and the ash bed in boiler#3 and #4
- various peep holes and inspection windows

The benefits of maintaining adequate level of excess air and flue gas temperature level through plugging these leakages and heat recovery are worked out in the table 2.2.9.1.2b. These are calculated on the basis of 4000 working hours per annum. The cost of coal is taken as Rs. 1200 per ton. Annexure 2.4 gives the detailed calculations.

Table 2.2.9.1.2b Improvement in efficiency

	Efficiency with air infiltration (%)	Efficiency without air infiltration (%)	Coal savings/ annum (MT)	*Monetary savings (lakh Rs)*
Boiler#2	75.31	81	170	20.5
Boiler#3	68.45	77.61	897	10.7
Boiler#4	68.73	77.84	818	9.8

2.2.9.2 Loss due to unburnt in ash

During the audit, problem of high unburnt in fly and bottom ash was observed in FBC and stoker fired boilers. To check the combustible levels in the ash, fly and bottom ash samples were collected and analysed in ACC laboratory. The combustible levels are reported to be 15.3% and 14.9% in fly and bottom ash of

FBC and stoker fired boilers respectively. Higher unburnt in fly ash could be due to

- high coal fines and overfeed system in FBC
- uneven coal sizing in stoker fired boilers

It was reported that the boilers are designed for 3-10 mm coal size. However, sieve analysis carried out during the visit, gives the following result (Table 2.2.9.2).

Table 2.2.9.2 Sieve analysis for FBC boiler in OCPP

Sieve size	+12.5mm	+10mm	+8mm	+6.3mm	+3.3mm	+2mm	+1mm	-1mm
Upper plenum	Nil	2.0	3.1	6.7	18.6	19.5	12.8	37.3
Lower plenum	Nil	0.4	6.0	14.8	12.6	21.5	12.2	32.2
Average	Nil	1.2	4.6	10.8	15.6	20.5	12.5	34.8

It is clear from the above table that more than 65% the coal is smaller than 3.3 mm size and about 35% is less the 1 mm size. The particle size <1 mm is the main contributor to the higher combustible in the fly ash, from FBC boiler, which escape with flue gas. The fluidisation velocity is too high to burn these particles in the bed. Sources of fines could either be coal preparation unit or the coal consignment itself or both.

Another reason for high unburnt could be the overfeed firing system in FBC (Boiler#2) of old CPP. In under feed firing system, fine particles have lesser tendency to escape with flue gas. It is expected that under feed firing system in FBC will reduce unburnt level to about 5% in fly ash and, therefore, switching over to this system is recommended in boiler#2.

In the stocker-fired boilers (boiler#3 & #4) the coal size is very uneven. It varies from fines of 3mm to lump of about 10cm. Grits were also observed with the coal. Uneven sizing of the coal is one of the reasons for higher unburnt. Larger lumps and fines should be avoided. The coal size should preferably be in the range 1-3 cm for stoker firing.

Ash samples were collected during the boilers' performance test and sent to the KCW laboratory for determination of combustible. This loss due to combustible is 6.72 % for boiler#2 and 8.86% for boiler#3 & #4.

2.2.9.3 Loss due to hydrogen and moisture in fuel

This is another big loss and depends upon the moisture and hydrogen content of the fuel and flue gas exit temperature. The moisture and hydrogen content of the fuel is fixed and, therefore, the only controlling factor is the exit flue gas temperature. The loss due to hydrogen and moisture is worked out to be 6.66%, 6.72% and 6.56% in boiler#2, #3 and #4, respectively. This loss will be reduced to a level of 6.21%, 6.12% and 6.12% in boiler#2, #3 and #4 respectively after plugging the infiltration and maintaining the flue gas temperature 150°C.

2.2.9.4 Loss due to Moisture in air

This loss is because of moisture content in air, which depends on the relative humidity and the ambient temperature. The other controllable factors, on which this loss depends, are excess air level and flue gas temperature. Presently, this loss is calculated to be 0.35%, 0.49% and 0.47% in boiler#2, #3 and #4, respectively and it will be reduced to 0.19%, 0.23% and 0.22% in boiler#2, #3 and #4 respectively by maintaining the desired levels of excess air and flue gas temperature.

2.2.9.5 Radiation loss

The structural losses in both the boilers is calculated to be 0.5% which is well within the limits. However, few spots like inspection doors and windows need special attention as the temperature is too high there.

2.2.9.6 Sensible heat loss in ash

This heat loss is due to sensible heat carried away in bottom ash and fly ash and depends on the bottom ash disposal and flue gas temperature. These losses are 0.34% for boiler#2 and 0.26 for boiler#3 & #4.

2.2.10 Instrumentation

Instrumentation is poor for OCPP boilers. There are control panels for boiler #1 and #2 (FBC boilers), however there are almost no controls for stoker fired boilers. Steam flow meters and coal meters are installed for FBC boilers only but they do not function properly. Controlling of the draught is manual but it can be controlled from control room for boiler#1 & #2. No similar facilities were provided for stoker fired boilers. It is recommended to rectify all the available controlling instruments on the panel to have a better control on the FBC boilers. Similarly, it is also recommended to install some critical control instruments for stoker fired boilers as they are also operating sufficient numbers of days.

2.3 New CPP turbine

Coal consumption figures for the period April to September 2001 are given in table 2.3.a.

Table 2.3a Unit heat rate on the basis of coal consumption

Month	*Generation (lakh unit)*	*Generation loss due to steam dump (lakhs unit)*	*Coal consumption (kg/kWh)*	*Coal consumption after considering effect of dump steam*
April	132.60	6.93	0.820	0.77
May	139.96	7.73	0.814	0.70
June	150.63	7.23	0.887	0.84
July	138.91	12.17	0.971	0.89
August	131.90	8.38	0.926	0.87
September	136.85	9.52	0.859	0.80
Average coal consumption (kg/kWh)			**0.880**	**0.81**
Average heat rate (kCal/kWh)			**3432**	**3159**
(Assuming average GCV of coal = 3900)				

Table 2.3b gives unit heat rate values obtained from table 2.3a.

Table 2.3b Unit heat rate and coal consumption

	Max heat rate (kCal/kWh)	*Min heat rate (kCal/kWh)*	*Average heat rate (kCal/kWh)*	*Average coal consumption (kg/kWh)*
Basis of actual generation	3785	3174	3432	0.880
Considering effect of dump steam	3471	2730	3159	0.810
Difference	314	444	273	

The above table indicates that if we include the effect of dump steam, the unit heat rate comes down by about 273 kCal/kWh. It may also be observed that there is wide variation in the overall heat rates (to the extent of 611 kCal/kWh).

2.3.1 *Heat rate on the basis of turbine heat balance*

Heat rate obtained on the basis of turbine heat balance and boiler losses is 3277 kCal/kWh which correspond to 0.70 kg coal consumption having GCV of 4680 kCal/kg. It can be observed from the calculation below that turbine heat rate works out to 2601. Calculations have been worked out on the basis of figure supplied by plant engineer.

a. Turbine heat rate calculation

Steam flow at turbine inlet	-	107 tons/hr, 480°C 62 kg/cm^2
Specific enthalpy of M.S.	-	807.87 kCal/kWh
Input enthalpy	-	107 x 807.87 x 10^3 kCal = 86442 kCal
Work done by turbine	-	Input enthalpy – ethalpy extracted and given to feedwater by way of condensate + LPH + HPH + deaerator steam
Feedwater temperature	-	200°C
Feedwater enthalpy	-	107 x 200 x 10^3 = 21400 kCal
Enthalpy used in producing energy by turbine	-	86442 – 21400 = 65042 kCal
Energy produced	-	25000 kWh/hr
Heat rate	-	$\frac{65042}{25000}$ = 2601 kCal/kWh
b. Average Boiler efficiency has been worked out to be	-	$\frac{81.21 + 82.45}{2}$ = 81.825% (based on audit data)
c. Heat rate after taking into account boiler efficiency	-	$\frac{2601}{0.81825}$ = 3179 kCal
d. Heat rate after taking into account generator efficiency at 97%	-	$\frac{3179}{0.97}$ = 3277 kCal
Specific coal consumption (taking GCV at 4680 kCal/kg as measured during tests)	-	$\frac{3277}{4680}$ = 0.70 kg/kWh

A look at the coal consumption values in table 2.3a indicate that consumption in the month of May was 0.814 and in July 0.971 kg/kWh corresponding to almost same level of generation (139 lac) unit. There is no consistency in the consumption pattern all through April to September. In a stabilised plant, such variation can not be technically explained. It is strongly felt that coal measurements need double check.

The effect of dumped steam should be reflected in arriving at the actual value of heat rate. Table 2.3b indicate that coal consumption after adding energy that would have been generated but for dumping of steam. The average value for the period April to September is 0.81 kg/kWh corresponding to heat rate of 3159 kCal/kWh and GCV of coal 3900 kCal/kg.

Unit heat rate obtained on the basis of turbine heat rate and accounting for boiler and generator efficiency works out to 3277 kCal/kWh. This compares well with the average heat rate of 3159 obtained after taking into account the effect of dumped steam (Table 2.3b).

2.3.2 Turbine operation

On the turbine side, heat balance reveals the turbine operating at heat rate of 2601 kCal/kWh. It could not be compared with design value since these were not available. Operating parameters at 25 MW were as follows.

Steam flow - 107 ton
Exhaust hood temperature - 44.74
CW inlet temperature - 29.4
CW outlet temperature - 37.8
Vacuum - 0.9
Terminal temperature difference - 6.94
Inlet – outlet temperature difference - 8.4

Operating values are well within the accepted norms and therefore, no heat rate is lost on account of turbine/condenser operation.

2.4 Old CPP turbine

The turbines are Stal make of 6, 8.5, 6 MW capacity. The base year of installation of 6 MW and 8.5 MW turbine is 1950 and 1957 respectively. Third turbine of 6 MW was added in 1996 (old turbine brought from Lakhari).

The maintenance of each turbine is carried out in 3 years from the manufacturer (ABB).

The residual life assessment of all the boilers was carried out last year and were cleared for 10 years of additional life. RLA of turbines is to be carried out.

The condensers of 6 and 8.5 MW turbine have been retrofitted with new condenser and third one is under way. Thus, condition of boiler and turbine is in operatable state.

Only one turbine at part load was in operation with vacuum at 0.87 during the period 15 October to 19 October. The coal consumption per kWh comes to 1.25 kg thus giving a heat rate of 4900 kcal/kWh at GCV of coal being 3900 kcal/ Heat rate is high because of low boiler efficiency. The FBC has no provision for grit return and stocker boiler operate at 60% PLF. Operational records shows turbines running at good vacuum (0.89 kg) with inlet and outlet cooling water temperature difference of 6°C.

All the turbines are robust and can run without giving any problem. In order to improve plant efficiency, surplus steam from the two boilers of new CPP should be used.

3

Pumps and fans

3.1 Pumps

The plant has a number of pumps of different types and capacities. Centrifugal pumps are largely used for pumping water for different applications. Only larger sized pumps with electrical drives above 60 HP were targeted for detailed scrutiny, since the potential to conserve energy is significant in such pumps. A study was conducted to assess the performance of these pumps as regards their performance under the prevailing operating conditions. The specifications of the pump are given in Table 3.1.

Table 3.1 Specification of pumps

S No.	Equipment	*Capacity (m^3/hr)*	*Head (m water column)*	*Rated power (kW)*	*Rated voltage (kV)*
1.	Boiler feed pump	76.3	930	290	6.6
2.	Condensate extraction pump	111.3	149	75	0.415
3.	Circulating water pump	2800.0	19.76	175	6.6

3.1.1 Evaluation of pump performance

To evaluate the performance of pump, the parameters like flow and pressure were recorded from the meters installed on the pumps. The performance of pumps has been evaluated using the following method.

The efficiency of a pump is defined as the ratio of useful power output to the power input to the pump shaft.

$$\text{Pump efficiency} = \frac{\text{Power output}}{\text{Power input}}$$

The power output of a pump is the energy delivered to the fluid by the pump. The formula for its calculation is as below.

$$P_{out} = \frac{\rho Q g H}{1000} \text{ kW}$$

where,

ρ = Specific gravity in kg/m^3
Q = Flow in m^3/sec
g = Gravitational constant (9.81)
H = Head in meters

The actual power taken by the motor was measured with the help of an accurate digital power multi-meter and the power input to the pump-shaft was obtained after considering the efficiency of the motor.

3.1.2 *Observation, analysis and findings*

Based on the measurements and data collected, the efficiency of these pumps is evaluated. Summary of calculations is tabulated below.

Table 3.1.2 Pump efficiency

					Measured parameters		
S No.	Application	*Rated power (kW)*	*Power drawn (kW)*	*%motor loading*	*Flow (m^3/hr)*	*Head (m water column)*	*Efficiency (%)*
1.	Boiler feed pump	290	255	77.12	49.3	1050	61.2
2.	Condensate extraction pump	75	63.6	79.7	103.2	150	70.5
3.	Circulating water pump	175	173	93.9	--	--	--

3.1.2.1 Condensate extraction pump

The efficiency of the pump is found to be 70.5% which is fairly good. It is also observed that flow of the pump varies with load on the turbine. The flow control in the existing system is done by throttling. The hourly variation in flow was recorded for a day. The data were analysed to determine the feasibility of a variable speed control instead of a conventional throttling.

Following assumptions are made while calculating the savings with variable speed AC drives.

- The sample data for a day is taken to be representative of the loading pattern for the whole year.
- The pump operates for 8760 hours in a year.
- Reduction in speed will maintain the minimum pressure requirement.

The estimated annual savings are to the tune of 49,600 kWh (i.e. Rs 79,400/- per year) with simple payback period of 6.8 years. As long term option, this proposal can be taken up. The calculation of energy savings by installing variable speed drive for condensate extraction pump is as follows:

Rated power - 75 kW
Measured power - 63.6 kW
Rated flow - 111.3 m^3/hr
Present flow - 92 m^3/hr for 8 hours
1110.3 m^3/hr for 16 hours

By reducing speed by 10%, process flow requirement can be achieved.

Power savings by reduction in speed = $63.6 [1 - (0.9)^3] = 17$ kW

Annual power savings = 17 x 8 x 365
= 49,640 kWh
Annual cost savings (@ Rs 1.6/kWh) = Rs 79,424/-

Investment required
Variable speed drive cost for 75 kW = Rs 5,41,000
Simple payback period = $\frac{5,41,000}{79,424}$
= 6.8 years

3.1.2.2 Boiler feed pump

Each boiler is connected with one boiler feed pump of capacity 76 m^3/hr. The normal operating load on the pump is 35 m^3/hr minimum and 50 m^3/hr maximum. So, the feed water pump runs underloaded. The efficiency of the pump at 50 m^3/hr is found to be 61.2%. Data of hourly variation in flow was collected to explore the feasibility of using variable speed drive. It was found that though there is load variation on the turbine, the same is not reflected in the boiler feed pump because of steam dumping. Moreover, the motor driving the boiler feed pump is in medium voltage range (6.6 kV). This would require MV drives. Medium voltage drives are costly. Though manufacturers claim they can provide solutions in MV area, we have not come across implementation experiences of MV drives particularly for energy saving applications in pumps and fans. However, there are successful international implementation experiences in the pump and fan applications. It would not be economical to try out these technologies in such applications.

3.1.2.3 Circulating water pump

Circulating water pump supplies cooling water continuously. The pump efficiency could not be evaluated, as the flow is not measured. The motor is found to be 93.9% loaded which is fairly good. The flow of circulating water pump will not vary much with load on the turbine because of steam dumping. Secondly, the condenser vacuums could be effected if flows are reduced beyond permissible limits. This, therefore, imposes restrictions in terms of a minimum flow in the system. Moreover, the motor driving the circulating water pump is in medium voltage range.

3.2 Fans

A study was conducted to assess the performance of ID/FD fans as regards their performance under the prevailing operations conditions. Only larger sized fans where flow control is done by controlling the outlet damper position were targeted for detailed scrutiny, since the potential to conserve energy is significant in such fans. ID and FD fans installed in the new CPP already have DC variable speed drives for flow control. Therefore, they were not targeted for the study.

3.2.1 Capacity control

Fan output can be controlled using outlet dampers, inlet dampers, variable vanes, variable pitch blades, variable speed prime movers, or by operating the fans in series or parallel. The system efficiency differs for each of these options.

The system curve is a simple function that can be stated as

$$P = K * (CFM)^2$$

Where,

P = pressure required to produce a given flow in the system

K = system constant and represents the friction to air flow

CFM = desired airflow

In the outlet damper control method, the flow through the system is changed with the change in the damper position. The outlet damper method affects the K value of the system curve. It is synonymous to adding resistance to the system circuit. The outlet dampers do not modify the fan curve but modifies the system curve as per requirement. This method does reduce the power input to the prime mover with the reduction in the flow, but this is the least efficient amongst all the methods of flow control. Thus this method of flow control is very wasteful of power and highly inefficient.

More efficient of all the methods for speed control is variable speed drives. It involves reducing the speed of fan to meet reduced flow requirements. Since the power input to the fan changes as the cube of the flow, this will usually be the most efficient from of capacity control. Another advantage of the variable speed control is the ability to provide step-less flow control.

This method takes advantage of the change in the fan curve that occurs with the change in speed of the fan. These changes are quantified in a set of formulas called the affinity laws. These laws are as follows.

2.2 *Affinity laws*

$Q_2/Q_1 = N_2/N_1$

$P_2/P_1 = (N_2/N_1)^2$

$HP_2/HP_1 = (N_2/N_1)^3$

Where,

Q = Flow (CFM)

P = Pressure (State inches of water)

N = Fan speed

HP = Horsepower

From the above laws, we get an equation

$$P_2 = [P_1/(Q_1)^2] * (Q_2)^2$$

The term $[P_1/(Q_1)^2]$ represents the system constant, K. This signifies that the fan will follow the system curve when its speed is changed.

2.3 *Fan performance testing*

Evaluation of fan performance requires various measurements of flow, pressure and power. It is essential that these parameters are measured simultaneously and there is no variation of load. The total efficiency of the fan is defined as

$$\eta_t = [(Q_{vi} * P_t)/W_i] * 100$$

where,

Q_{vi} = Inlet volume flow (m^3/s)

P_t = Fan total pressure (Pa)

Wi = Impeller power (Watts)

Since the impeller power is not easily measured so it is recommended to measure power input to the prime mover (electric motor in this case). The equation given above will represent the gross efficiency of the fan system. The equation is rewritten as

$$\eta_g = (Q_{v1} * P_t)/W_e * 100$$

where,

W_e = Electrical power input to the motor (watts)

η_g = Gross efficiency of the fan system

Now, fan total efficiency is defined as

$$\eta_t = \eta_g/(\eta_m \cdot \eta_d)$$

where,

η_m = Efficiency of the motor

η_d = Efficiency of the drive

Since these two efficiency figures are easily available from the manufacturers, thus, the total fan efficiency can be determined.

The above formula holds good for pressures upto 2500 Pa when the air can be considered incompressible. At higher pressures, it is necessary to take account the rise in the air density through the impeller. The air discharged will actually be compressed to a volume flow rate smaller than the previous one. Both the work done and the input power taken by the motor are reduced by these changes, and, to avoid overestimating the efficiency, the flow is reduced to mean value K_pQ_{v1}, where K_p is called the compressibility factor.

$$\eta_t = (K_p \cdot Q_{v1} * P_t)/W * 100$$

3.2.4 *Measurements and observations*

Measurement of air flow and pressure of ID and FD fans of stoker fired boilers were done. Mechanical efficiency of the fans is calculated and is given in Table 3.2.4.

Table 3.2.4 Mechanical efficiency of FD and ID fans

Fan application	Rated power (kW)	Power drawn (kW)	% loading motor	Designed discharge (m^3/s)	Measured discharge (m^3/s)	Measured pressure (mmWG)	Fan efficiency (%)
FD fan	45.0	25.8	52.7	17.3	10.7	101.7	45
ID fan	67.5	44.8	60.3	--	27.2	91.0	60

It can be seen from the table that the efficiency of the fans is low because they are running between 50-60% loading.

3.2.5 *Installation of variable speed drive for stoker fired boiler FD fans*

There are two stoker-fired boilers of 20 tph capacity each generating steam at 30 kg/cm^2. Each boiler is provided with one FD fan. The damper position of both the FD fans is kept at 75% opening. The boilers are capable of generating 4 MW of power but due to poor quality of coal available, the boilers generate maximum of 2.5 MW power. The load on the boilers does not vary much. By opening damper position 100% and reducing speed of fan, 10-15% energy savings is possible. Installing variable speed drive and reducing fan speed by 10%, the estimated savings to the tune of 35,000 kWh per year (5066 hrs/year) for one fan amounting to Rs 85,000 (@ Rs 2.4 per kWh) can be achieved with an investment of about Rs 4,65,000 yielding a simple payback period of 5.4 years. However, discussion with plant personnel revealed that they are planning to convert the stoker-fired boilers to FBC in near future. In that case, it is recommended that use of variable speed drive for FD/ID fans should not be considered at present because the economics as well as configuration of the drive may change after conversion to FBC boiler.

3.2.6 *Installation of variable speed drive for FBC boiler FD/ID fans*

There are two FBC boilers of 25 tph capacity each generating steam at 31 kg/cm^2. Each boiler is provided with two FD fans and one ID fan. Each boiler is having two numbers plenum (lower and upper) with air nozzles. The FD fans damper position is kept fixed at 50% opening with two plenums in operation. Discussion with plant personnel revealed that they do not change position of damper with load as there is a requirement of minimum fluidisation velocity (bed pressure 460 mm water column) with boiler. And in case the load is less than 50%, they slump one bed. This is an inefficient way of operation and results in high excess air level at low load as mentioned in Chapter 2.

In absence of characteristic curves and system resistance curve (including the fluidisation velocity) it is not possible to comment on the feasibility of using variable speed drive in this application.

4

Compressed Air Systems

4.1 Introduction

There are nine reciprocating type air compressors operating in the captive power plant (CPP). The number of compressors in the old and new powerhouse is 2 and 7 respectively. Three (3) and four (4) numbers of air compressors are operating for FBC boiler instruments and conveying (ash handling) systems in the new powerhouse. The compressors in old and new powerhouse are lubricated, air cooled type and oil-free non-lubricated, water-cooled type. Both the compressors in old powerhouse (OCPP) are of two stages. All the compressors in new powerhouse (NCPP) are of single stage. The number of compressors in running condition in NCPP depends on the operation of pneumatic systems/ sub-systems. Normally one compressor in each powerhouse is stand by.

The compressed air is generated and supplied for various purposes for a number of pneumatic operations and controls in the various power generating equipment through number of receivers and headers. Two receivers (out of room) with one common header and three receivers with one common sub-header (within room) are operating in OCPP and NCPP. Two receivers in OCPP are interconnected through a valve. Each compressor is fitted with one cylindrical, drum-type suction air filter separately. The installation of one air drying unit has been completed, but not taken in operation for conditioning (removal of moisture and dirt) of compressed air. Three receivers in ash handling units of NCPP are suitably interconnected to obtain air through different discharge pipes, common header and any of the four compressors. The pressure gauge is fitted with the individual compressor and receiver in both powerhouses. Each compressor is fitted with two (at left and right side) cylindrical, drum type suction air filter separately. Air receiver is fitted with manually operated drain valve to facilitate the water drainage.

4.2 Technical information

The average operating pressure at the receivers in OCPP is about 8.1 kg/ cm^2 (g). The operating pressure range at the receivers in ash handling units of NCPP is 3.6-3.8 kg/ cm^2 (g). The installed pressure gauge on compressors in OCPP is working with the zero error of +0.66 kg/ cm^2 (g). The in-built pressure gauge on compressor #3 and #4 in NCPP is also working with the zero error of +0.2 kg/ cm^2 (g). During testing the compressors in both powerhouses switched-on and off automatically as the lower and upper pressure limits were achieved. The compressors in NCPP are individually fitted with after-cooler using water. Both the compressors in OCPP will shortly be fitted with the deliquescent type dryer, placed out of the compressor room. The air dryer in operating condition will dry (free from moisture) all of the air discharged from the individual compressor, to reduce the dew point by 8-10°C. The designed operating pressure and temperature of air dryer would then be 9.5/8.8 kg/cm^2 (g) and 190/40-140°c respectively.

The details of six reciprocating compressors is given in Table 4.2a. The discharge pipeline of each compressor is fitted with common header.

Table 4.2a Detail of reciprocating type air compressors

Sl. No.	Specification no./ Manufr's sl. no.	*Make*	*Motor (kW)*	*Rated pressure (kg/cm^2)*	*FAD (m^3/min)*
	OCPP				
1	No.1-ESV-NL1/ 9640971	Ingersoll-Rand Co. Ltd.	18.5	7.7	N A
2	No.2-ESV-NL2/ 9650193	Do	18.5	7.7	N.A.
	NCPP				
3	1H X 2T-25/ N-3557008	Chicago Pneumatic Ltd.	55	3.7	10.2
4	Do	Do	55	3.7	10.2
5	Do	Do	55	3.7	10.2
6	Do	Do	55	3.7	10.2

N.A. - Not available

Plant personnel of OCPP reported that the continuously operating machines would require drying of compressed air. The broad areas of compressed air consumption in NCPP and OCPP are FBC boilers (new) instruments, ash handling units and FBC boiler (retrofitted) instruments. The collection of data (logbook and design) and monitoring of operational parameter for three compressors in NCPP with FBC boiler's instruments was not done because the individual compressor can't be isolated completely from instruments of the entire plant for any moment of time. There are two identical receivers and three identical receivers operating in the OCPP and NCPP respectively. Plant personnel provided the standard capacity/ volume of receivers and interconnecting pipes in NCPP as given in table4.2b.

Table 4.2b Capacity/volume of receivers and interconnecting pipes in NCPP

Sl. No	System/ component	*Capacity (m^3)*
1	Receiver	15.00
2	Interconnecting pipes of 200 NB	0.54
3	Interconnecting pipes of 100 NB	1.56
4	Interconnecting pipes of 80 NB	0.048
5	Interconnecting pipes of 65 NB	0.021*
6	Interconnecting pipes of 50 NB	0.023**

* connected with AHL 3, ** installed after receiver

The approximate length of interconnecting pipe size of 200 NB, 100 NB, 80 NB, 65 NB and 50 NB is 17.2 m, 198.75 m, 9.75 m, 6.6 m and 12.2 m respectively. The annual operating hours of compressor #2, #3, #4 in NCPP and #1 in OCPP is 5600, 5600, 5600 and 7560 respectively.

3 Free air delivery (FAD) test

Compressors are designed to deliver the fixed quantity of air at certain pressure. But, due to ageing, wear and tear or poor maintenance, compressor may not deliver the specified quantity of air. By performing this test, actual output of a compressor can be assessed. Performance of the four reciprocating compressors (No. #2, #3, #4 in NCPP and No. #1 connected with vessel/ receiver 2 in OCPP) is given in table 4.3.

Table 4.3 Operating parameter of receiver and compressors

Sl.		*OCPP*	*NCPP*		
No	Parameter	*Comp. #1*	*Comp.#2*	*Comp.#3*	*Comp.#4*
1	Capacity of the receiver (m^3)	0.49	15.0	30.0*	15.0
2	Capacity of the interconnecting pipes (m^3)	0.015	2.171	2.171	2.171
3	Initial pressure (bar abs.)	1.673	1.013	1.013	1.013
4	Final pressure (bar abs.)	4.85**	3.53	3.53	3.53
5	Pump-up time (minutes)	1.72	6.00	8.43	9.07
6	Actual FAD (m^3/min)	0.92	7.11	9.49	4.71
7	Design FAD (m^3/min)	N.A.	10.2	10.2	10.2
8	Average of actual load on motor (kW)	3.31	9.70	9.76	9.60
9	Sp. Power consumption (kW/m^3/min)	3.60	1.36	1.03	2.04

* two receivers were interconnected, **after eliminating zero error of pressure gauge, N.A. – Not available

If the actual free air delivery, by the above test, comes below 80%, of the rated or designed capacity, the compressor needs overhauling. It is clear from the table that actual FADs of compressor #2, #3 and #4 in NCPP are 69.7%, 93.0%, and 46.1% respectively. The FAD of compressor #1 in OCPP is 0.92 m^3/min. The actual free air delivery could not be compared with the designed parameter, because it is not available. The FAD of compressor #3 is okay where as the same of compressor #2 and #4 is below the satisfactory level. Because of this specific power consumption for compressor #2 and #4 is quite high compared to compressor #3. It is, therefore, recommended to get compressor #2 and #4 in NCPP overhauled to have the better productivity and lower energy consumption. It is also recommended to contact the manufacturer for the designed FAD of compressors in OCPP. This can be compared with the measured value and then decision taken regarding overhauling.

4.4 Compressed air leakages

Compressed air leakage forms a major source of energy wastage. During the energy audit study, leakage was noticed from selective pipe joints, valves pneumatic tools, etc. It is strongly recommended to identify the leakage points and arrest the same. Though, it is not possible to make compressed air system 100% leak proof, but air leakage above 5%, certainly, need in-depth study of the network. The leakage test, to quantify the leakages in the distribution lines of ash handling system 2 and 3 of NCPP was carried out. It could not be carried out in the distribution system of OCPP because operation of one FBC boiler's instruments were using compressed air continuously, through out the period during the visit.

The leakage of compressed air in the ash handling lines (AHL) can be quantified by running the compressor with all air using equipment shut off. The leakage test for both AHLs was carried out with the compressor #2. Three sets of reading for individual AHL was taken in the NCPP. The leakage rate in the compressed air line 2 and 3 is calculated to be 9.5% and 6.7% respectively. Even leaving a margin of 5.0% leakage for untraceable points in the compressed air line, the reduction in electricity consumption achievable in the unit, with plugging out the leakage would be reasonable amount annually. The annual electrical energy saving is 2430 kWh and 940 kWh, and the corresponding monetary saving would be Rs. 4900 and Rs. 1900 respectively. The details are given in annexure 4.1.

4.1 Detecting leakage points

It is difficult to detect air leakage as they can not be seen and smelt. While large leakages are easily detected by hissing sound generated, it is difficult to detect small leakages, which can only be detected by applying soap solution. Some of the most susceptible points are:

a) threaded pipe joints
b) flange connections
c) valve stems
d) filters
e) hoses
f) connectors
g) operating valves on pneumatic devices
h) check valves
i) relief valves
j) end use machines

The compressed air leakage detection test was carried out in three AHLs. Minor loss of compressed air was observed during the different set of tests in few valves in AHL 1 and 3.

The air loss due to leakage can be quite significant depending on the air pressure. Table 4.4.1 gives the leakage through various orifice size and energy wastage at 7 kg/cm^2 (g) air pressure.

Table 4.4.1 Power wastage at 7 kg/cm^2 (g)

Orifice dia	*Air leakage (scfm)*	*Power wasted (kW)*
1/64"	0.406	0.08
1/32"	1.62	0.31
1/16"	6.49	1.26
1/8"	26	5.04
1/4"	104	20.19

(m^3/min = 39.2 scfm)

Plug the leakages immediately after detection. It is recommended to maintain properly the compressed air distribution system at different AHL and get the entire system tested with soap solution once in a year.

4.5 Optimisation of compressed air system

Compressed air from the water-cooled reciprocating compressors is available for main instruments, associated components in three AHL and receiver & interconnecting pipes. The sequence of equipments has been arranged to perform the following functions.

1. Removal of particulates as small as three microns with minimum pressure loss using inlet air filters.
2. Reduction of air temperature and removal of moisture (liquid water) using an intercooler and a separator.
3. Reduction of air pressure pulsation and moisture carry-over by using a receiver.

There is no aftercooler with the individual compressor in OCPP. The conventional type compressed air dryer after the receiver in OCPP should be in working condition for the removal of water vapour and liquid in the line.

The energy efficiency of a compressed air system can be optimised by maintaining/ installing equipments/ components for proper function. All of the systems/ components in the compressed air distribution system should be maintained as per the manufacturer's recommendations. Following measures are recommended to improve the performance of the compressed air system.

- Operate the air dryer properly in OCPP to prevent moisture from condensing in the distribution system and to have a lower dew point.
- Plug the air leakage. Most of the threaded joints in the pipes on ground/ overhead should be made leak proof by welding at the ends.
- Operate the compressor #2 and #4 in the NCPP at full load condition.

Normally, the energy efficiency of refrigerated type air dryer is higher than the deliquescent type air dryer. The refrigerated type dryer are installed to dry (free from moisture) all of the air discharged from the individual compressor by cooling the air from dew point of 1-4°C to about -15°C.

4.6 Inlet air temperature

Two very small sized air intake pipes and filter at two ends are located adjacent to the individual compressor in NCPP. Four compressors and associated components are located within the compressor room. Temperature of air in the ambient, compressor room and the individual suction air was measured using thermocouple and it was found that the temperature of the inlet air to the compressor in OCPP was higher than the ambient temperature. The actual temperature of suction (inside room) air for four compressors has been given in the table 4.6a.

Table 4.6a Air temperature for suction (inside room) of compressors

	OCPP	*NCPP*		
Parameter	*Comp #1*	*Comp #2*	*Comp.#3*	*Comp.#4*
Ambient temp (°C)	30	30	30	30
Suction air temp (°C)	33.1	30.4	30.6	30.5
Difference (°C)	3.1	0.4	0.6	0.5

The actual temperature of suction air in compressor #1 is above the ambient temperature. The actual temperature of suction air in compressor # 2, #3 & # 4 is close to the ambient temperature. However, care should be taken against any increase in the suction air temperatures. The inlet temperature of air to the compressor is very important, as rise in temperature will increase the energy consumption because of increased volume of air. Thus, lesser the temperature of air at compressor inlet, lesser would be the energy consumption of compressor. The effect of inlet air temperature on compressor performance is given in Table 4.6b.

Table 4.6b Effect of inlet air temperature on compressor performance

S.No	Temperature of intake air (°C)	*Volume required to deliver 1000 m^3 at 21°C (m)*	*Percentage saving/loss related to 21°C*
1	5	943	5.7
2	10	982	3.8
3	16	981	1.9
4	21	1000	0
5	27	1020	-1.9
6	32	1040	-3.8
7	37	1060	-5.7
8	43	1080	-7.6
9	49	1100	-9.5

Note: The negative sign in the table indicates energy loss

Following measures are recommended for the performance improvement of the compressors.

1. Temperature of the air supplied to the suction of compressor #1 in OCPP should be monitored periodically to ensure that it is, at least, equal to the ambient temperature. In case any rise is observed, the inlet air duct can be extended further to outside of room.

2. Suction air to be maintained at the temperature close to ambient throughout the operation in all seasons for compressor #2, #3, #4 in NCPP and #1 in OCPP.

4.7 Use of automatic drain valve in air receiver

It was observed that five (3 number in NCPP and 2 number in OCPP) air receivers drain valve was manually operated to facilitate the water drainage. The drain valve of three air receivers for AHL in NCPP are provided with the bypass line arrangement. The drain valve of two air receivers in OCPP are not provided with the bypass arrangement. Moisture is drained from the five receivers once in a day. This provision not only passes the water but also allow the compressed air to pass to the atmosphere intermittently. By using the automatic drain valve, water would only be allowed to pass intermittently, thereby, reducing the wastage of compressed air. The addresses of the above drain valve suppliers are given below.

1. Trident Electric Pvt. Ltd.
   #133, I Floor, 11th Cross
   Opp. Jeerige Building
   Malleswaram,
   Bangalore-3

2. Siemag Hi-tech Filters Pvt. Ltd.
   #7, New Lata Apartments
   Jawahar Nagar, Goregaon(W)
   Bombay-40006

5

Cooling towers

5.1 Introduction

The captive power plant of the cement manufacturing complex is equipped with two large sized cooling towers. The old cooling tower in old CPP was replaced few years back by the new cooling tower at different location. Both the existing cooling towers are operating to meet the cooling water requirements of the old and new CPPs. The cooling towers (CT #1 in OCPP and CT #2 in NCPP) are counter flow type having 3-4 cells, each fitted with aluminium fan blades. Plant management reported that one set of fibre reinforced plastic (FRP) blades have been procured for the replacement of aluminium blades of one fan in NCPP. Both the cooling towers are of similar design, different model shower induced draft type with wall using asbestos sheet at two ends (left and right), MS fills vertically at the inside through which water falls in to the sump. It allows uniform, smooth airflow and water fall through identical passages at different height. The two sides (front and back) of the entire structure are made by the number of asbestos louvers/ baffles. The wall at two ends of each cooling tower prevents the airflow over the water surface, but it is essential for the stability of the structure. The uniform water distribution and uniform air flow are two major elements required for efficient cooling tower operation. M/s Paharpur Cooling Towers Ltd. have carried out commissioning of two cooling towers.

5.2 Technical details

The designed heat load (kcal/ hr) per cell of the cooling tower is not available. The return header of cooling tower delivers hot water to the cells where it is distributed on two sides of cell. The quantity of water for distribution can be controlled using properly sized butterfly valves. The detail of the cooling tower is given in the Table 5.2.

Table 5.2 Design detail of cooling towers (CT)

Parameter	CT #1 in OCPP	CT #2 in NCPP
Serving plants/ areas or source	Air compressors + Condenser	Condenser + Generator cooler, oil + Air compressors
No. of CTs/ No. of cells	1/ 4	1/ 3
Designed circulation rate of water	1900 m^3/hr per chamber	2600 m^3/hr per chamber
No. of pumps/ motors installed	Pump/ motor - 3/ 3,	Pump/ motor - 3/ 3,
Rating of pump (m^3/hr), motor (kW)	Pump - 1900, Motor - 165	Pump - 2800, Motor - 150.5
Dsg. head of pump (m)	24.0	19.67
Dsg. inlet water temp. (°C)	46	43
Dsg. outlet water temp. (°C)	35	33
Dsg. cooling range (°C)	N. A.	N. A.
Dsg approach (°C)	N. A.	N. A.
Dsg. pH limit of circulating water	N. A.	N. A

N. A. - Not available

5.3 Observations

The water spray patterns produced by all nozzles in the selective cell(s) of tower was observed while cell(s) are operating. All the patterns were not symmetrical and not equal between nozzles. Each cell is equipped with one properly sized sump. The individual pump is supplying cold water from the respective sump. The discharge pipes of number of pumps are interconnected in such a way that any operating cell can be used for the cooling of warm water from any area/ system in the plant.

The selective fans are manually switched off/ on during the plant's operation without/ with load in winter and other seasons. The operating hours of CT/ cell depends on the load on the cement kilns. There is no provision to monitor the fans operation automatically in cooling tower/ cell with respect to ambient conditions during the winter (3-4 months) season. Following are the observations made on two CTs separately:

CT #1 - The average temperature of water at left and right side of return header (inlet to CT) is 45.15°C and 45.4°C respectively. During the time of the energy audit, it was found that the second (from the front gate side) cell was in operation condition. Large quantity water was directly falling at the opposite side of pumps, because most of the nozzles and splash reverse cone type holes in that segment are choked. The average temperature of water at left, right side of CT/ cell's base and sump (outlet of CT #1) is 36.65°C, 33.5°C and 33.02°C, 37.5°C, 32.1°C, 35.9°C respectively. Normally, one pump is stand by. The pressure in installed gauge at the supply side of pump (outlet of CT & 2nd pump) is 1.3 kg/cm^2. The suction head is 1.5 metres.

CT #2 – The cold water from this cooling tower is supplied to TG auxiliary cooler, Sample cooler, Air compressor, Journal bearing cooler and Ash drain cooler. The temperature of water (from five areas) at the return header (inlet to CT) is in the range 36.1-36.3°C. During the time of the energy audit, it was found that the second (from the entry side) cell was in operating condition. The temperature of water at the cell's base (outlet of CT/ cell) and sump is in the range 28.6-29.1°C and 23.5-25.3°C. Normally, one pump is stand by. The pressure in the installed gauge at supply side of pump (outlet of CT & 2nd pump) is 1.5 kg/cm^2. The suction head is 3.5 metres.

5.4 Performance Evaluation

The performance was evaluated when the selective cell of the cooling tower coupled with equipments operating continuously at normal load condition. Plant personnel could not provide the characteristic curves of circulating water pump. The designed heat load per cell/ chamber of the cooling tower #1 and #2 is 20.9 GCal/hr and 26.0 GCal/hr respectively. The actual heat discharge (kCal/ hr) per cell of the cooling tower could not estimated.

During the time of the energy audit, it was found that the remaining cells in each cooling tower were not in operating condition. There is no flow meter for the measurement of actual quantity of water flow. The make-up water is rarely (added) drawn from the treated water tank. The desired quantity make-up (corresponding to evaporation loss) water is added in supply sump. There is no automatically controlled valve for drawing desired quantity of make-up water. The range of make-up water required (m^3/hr) in two cooling towers is not available. The designed cooling capacity of two towers is different. The parameters (3-4 sets) measured were temperature of incoming and out going cooling water, ambient dry and wet bulb temperature, total dissolved solids, pH of water at cooling towers sump, etc.

The range (difference in cooling tower inlet and outlet water temperature) and approach (difference in cooling tower water outlet and ambient air wet bulb temperature) for two cooling towers are given in Table 5.2 along with the measured parameters. Range and approach are two main factors governing the performance of a cooling tower. For optimum performance, range should be as high as possible and approach should be as low as possible.

Table 5.4 Operating parameter and performance of cooling towers

Parameters	*CT #1*	*CT #2*
Operating cells (nos. and position)	1 (B)	1 (B)
Average in-coming water temperature to CT (°C)	45.3	36.2
Average out-going water temperature from CT/ in sumpm (°C)	35.0/34.0	28.85/24.4
Avg. pressure in installed gauge at supply pump/ outlet of CT (kg/cm^2)	1.3	1.5
Head (metres)	1.5	3.5
Dry bulb (ambient) temperature (°C)	27.4	27.4
Wet bulb (ambient) temperature (°C)	19.7	19.7
Actual pH limit of circulating water		
Range (°C)	10.3	7.35
Approach (°C)	7 6	1 45

N.A - Not available

The deterioration in water quality would increase the impact of fouling and scale, both of which rapidly decrease the performance and efficiency of the cooling tower. Pneumatic controlled float valve in sump should be installed for drawing desired quantity make-up (corresponding to evaporation loss) water in supply sump.

The incoming/outgoing water of cooling tower in #1 is at higher temperature than in #2. The make-up water (at about 40°C) is also mixed with the outgoing cold water in the sump of the cooling towers. Due to this, the average temperature of water in the sump of cooling tower #1 and #2 is 34.0°C and 24.4°C respectively. The temperature of water within fills/ louvers (before the sump) at different height was varying along the height and length of cooling tower #1. The approach is also higher in cooling tower #1. However, the outgoing water temperature in cooling tower #2 is close to ambient wet bulb temperature and, therefore, has the lower approach. The range is also lower because of insufficient cooling of incoming water through the fills/ louvers of the cooling tower #2.

The best performance of cooling tower is achieved by cumulative improvement of its components - the fill, distribution system, drift eliminators and fans.

5.5 Fills/ Louvers

The fills/ louvers in a cooling tower facilitate heat and mass transfer between the circulating water and the air. The construction of cooling towers is made with asbestos sheet. Plant personnel reported that all the cells are having MS fills inside, which perform uniformly. Moreover, the fills/louvers were partly bypassed in cooling tower #1 because of clogging and water was falling, directly in the sump and because of that non-uniform water falling temperature at the bottom of the fills was varying along the length of the sump. It is recommended to replace the clogged/jammed section of fills/louvers with new section, which provides maximum water exposed surface area to air for better heat transfer.

In cooling towers, the fills consisted of a series of splash bars that caused the water to fall turbulently, continually exposing new surfaces to the air flow. In both tower designs, if the tower's fill material is damaged, out of place, missing, or badly scaled, the surface area of the water that will be exposed to the air flow through the tower will be reduced, reducing tower performance.

5.6 Water distribution system

The performance of cooling tower depends on the operation and condition of its water distribution system also. The water entering the tower is evenly distributed as it enters the fills. Ideally, the cooling tower requires proper balancing of the water distribution lines and proper operation of the distribution nozzles. The water distribution system use butterfly (balancing) valves to regulate the flow through the distribution system, resulting in equal water flow into all tower cells. Large amount of water was falling in the cooling tower #1 because of choking of few nozzles. It is recommended to either clean the choked nozzles or replace the damaged nozzles.

5.7 Drift eliminators

As air flows through the tower, particularly, when the tower fans are operating at high speed, some of the circulating water gets entrained as droplets in the air flow and is carried off. This process is drift, which cause water loss and increase make-up water and water treatment chemical requirements. A separate component called drift eliminator, which is used to reduce the amount of drift released from a tower. Ensure proper maintenance of the drift eliminator.

5.8 Use of hollow FRP fan blades

One set of compact FRP blades is to be installed as the replacement of aluminium blades of one fan in NCPP. The blades of other fans in cooling tower #1 and #2 are made of aluminium. Nowadays, hollow FRP blades are available which can replace aluminium fan blades and contribute the energy savings. Hollow FRP blades being lighter in weight give higher efficiency. The higher efficiency is also due to the special aerodynamic design, streamline finish and lightweight of the blades. Also, the airflow rate through fans would improve by 5 to 10% and will yield energy saving of 15%.

By replacing the existing fan blades with hollow FRP in the cooling tower #1 and #2, a saving of 29330 kWh and 28670 kWh or Rs. 58,700 per annum and Rs. 57,400 per annum is possible as calculated in Annexure 5.1. The estimated payback period of would be around 9-15 months. The actual payback period will be calculated based on the firm quotation from the suppliers. Normally, the payback period for the replacement of aluminium fan blades with hollow FRP blades in cooling towers is attractive. The address of the supplier is given below.

1. Cooling Tower Corporation
   GIDC, Makarpura,
   Vadodara - 390 010
2. Melfrand Engineers Pvt. Ltd.
   Govind Nagar, Opposite NSS Road
   Home Guard Centre, Ghatkopar (W)
   Mumbai - 400 084

5.9 Thermostatic controller

The outgoing water temperature from the cooling tower fitted with CT #1 and CT #2 is 35°C and 28.85°C respectively. The designed setting of water temperature at inlet/ outlet of the cooling tower #1 and #2 is 46°C / 35°C and 43°C / 33°C respectively. It is suggested to switch off the cooling tower fan when the ambient temperature falls below the 28°C (during the months of December & January, 15 days of February & November and night of next 30 days in each of period February-March & October-November). It is proposed to install the simple electronic CT-ENSAVER+ system in each cell consisting of one micro-controller, two sensors. The temperature setting of controller with cooling tower #1 and #2 should be adjusted at 27-32°C. This retrofit will switch off the fan as soon as the water (in cooling tower sump) temperature falls below 27°C (duration is specified above) and switch on the fan at 32°C automatically. It also indicates cumulative fan on and off-hours to estimate the excess capacity of the cooling tower. During the fan off condition, only free fall area of cooling tower will be utilised. It is recommended to install the CT-ENSAVER+ system (thermostatic controller) in the cooling tower #1 and #2 fitted with OPH and NPH respectively. The savings in the cooling tower #1 and #2 because of this will be 59996 kWh and 58706 kWh or Rs.1.19 lacs per annum and Rs.1.17 lacs per annum as calculated for the cooling towers is given in the Table A5.2 in Annexure 5.2. The payback period for the installation of CT-ENSAVER+ system in cooling tower #1 and #2 is attractive. The address of the supplier is given below.

Ensave Systems Private Limited
3, Anand Shopping Center,
Second Floor, Bhattha, Paldi,
AHMEDABAD - 380 007
Phone : (079) 662-1116 / 661-4609
Fax : (079) 663-7907
E-mail : maxprotech@satyam.net.in

6

Coal handling system

.1 Introduction

New CPP coal handling plant has single belt conveying system with provision to sieve the coal through 3 mm screen and also down to 10 mm size at the sieving house. One granulator crusher with a capacity of 185 ton per hour has been provided. The crushed coal finally feeds coal hoppers having storage capacity of 500 tons (each boiler). Both the hoppers thus provide coal reserve for 48 hours (coal consumption at full load of 25 MW is 500 ton).

The fine coal obtained after the crusher below 3 mm size is far more in quantity than the value for which boiler has been designed.

The inspection of coal quality in the coal yard revealed that quantum of coal fines is very high. The new CPP therefore is bound to face the following problems.

1. Higher percentage of unburnt carbon in the fly ash. It is already around 15%.
2. Choking of hoppers and crushers during rainy season.
3. Coal feeding to the plant can become critical because of single stream. Choking of coal in bunkers, maintenance of conveyors, motors, crushers and screens can effect coal feeding to bunkers. These problems multiply during rainy seasons.
4. It is observed that present feed rate is 50 tons/hr as against belt capacity of 140 ton/hr. CHP has to be run for two shifts. If the feed rate improve to 80-100 tons, coal feeding can be accomplished in one shift.

The above observations need urgent attention so that power generation from new plant is uninterrupted all through the years. Coal quality, coal storage, coal conveyor, coal screening and belt capacity need urgent review. We suggest a separate storage for the appropriate size of coal for 25 MW unit. Table 6.1 gives the coal composition for July, August and September, 2001.

Table 6.1 Monthly average of sieve analysis: Ash, total moisture, GCV and combustibles

Sieve analysis	*July*	*August*	*September*
+ 8 mm	1.63	1.27	2.43
+ 6.3 mm	7.45	7.93	10.77
+ 3.3 mm	24.77	22.63	23.35
+ 2 mm	22.48	20.95	17.33
+ 1 mm	17.05	16.30	14.85
- 1 mm	26.42	30.9	31.42
Ash	33.57	32.40	33.60
Total moisture	14.37	14.10	13.10
GCV (kcal/kg)	3860	3963	3924
Combustibles	14.40	12.20	13.70

7 Water chemistry

7.1 Introduction

Raw water is obtained from the abandoned colliery. The quality of raw water is given in Table 7.1[✦].

Table 7.1 Quality of raw water

Parameters	
Turbidity and suspended solids	0.95 ppm
pH	8.2
TDS	227.5 ppm
Total hardness as $CaCO_3$	105.5 ppm
Calcium hardness as $CaCO_3$	20.00 ppm
Magnesium hardness as $CaCO_3$	13.49 ppm
Chlorides as Cl	30 ppm
Sulphate as SO_4	2.16 ppm
Silica as $S1O_2$	15 ppm
Conductivity	350 Mho/cm

7.1.1 Water treatment

Pretreatment is carried out by passing the raw water through sand filters to obtain clear water. This water is softened which is used for de-mineralized (DM) water and make up for the cooling tower loss.

DM plant comprises of two steams of 8 m^3 and 9.5 m^3 per hr capacities. The sequence is cataion, degaser, anion, and mixed bed. The capacity and quality of water at various stages is given in table 7.1.1.

[✦] From project report

Table 7.1.1 Capacity and quality of water at various stages

		Capacity
Pretreatment	-	62 5 x 3 m^3
Soft water	-	90 x 2 m^3/hr
DM water	-	8 + 9 5 m^3/hr
Quality		
Soft water		
Hardness	-	5 ppm
DM water		
Conductivity	-	0 2 Mho/cum
Silica	-	0 02 ppm
pH	-	7 8
Water consumption		
New plant	-	75 m^3/day
Old plant	-	120 m^3/day
Make up consumption	-	5%
Blow down	-	Continuous crack open
Boiler water regimes		
Silica	-	5 ppm
PO_4	-	10-15 ppm
pH	-	9 5 - 10 2
Steam side		
Silica	-	0 02 ppm
Conductivity	-	0 02 ppm
pH	-	9 5
Feed water		
O_2 content	-	0 01 ppm
Cost of DM water	-	Rs 20/m^3

7.1.2 Recommendations

- We recommend intermittent blow down instead of continuous blow down.
- Dosing of anti-corrosion chemical should be reviewed by assessing the efficacy of the chemical added.

Plant is being provided with conductivity meter in the main steam line. This will help to monitor the quality of main steam at the end use.

8

Energy Analysis

.1 Introduction

Keymore Cement Works (KCW) has two Captive Power Plants (CPP), namely old CPP and new CPP. Old CPP has three TG sets of 6, 8.5 and 6 MW capacity while new plant has one TG set of 25 MW. The average requirement of KCW is 35-36 MW including auxiliary power consumption. Details of load distribution on CPP is given in table 8.1.

Table 8.1 Load distribution on CPP

S No.	Plant	*Area*	*Load (MW)*
1.	New CPP	2 raw mills and one kiln	18.5
		Packing house load of 1 MTPA plant	4.5
	Sub-total		*23.0*
2.	Old CPP	Balance load of 1 MTPA plant	4.0
		Cement mills	5.0
	Sub-total		*9.0*
3.	Either new CPP or old CPP	Refractory plant, colony, factory lighting, workshop, water works, Keymore Engineering Institute	3.0
	Total		*35.0*

The plant manager brought out the following problems in the plant.

- High auxiliary power consumption
- High percentage of combustible in fly ash
- High heat rate
- Improvement of power factor beyond 0.88

In addition to these problems, the following issues have also been analysed and report given in the following sections of this chapter.

- Boiler operation at low loads
- Integrated operation of new and old CPPs
- Operation management
- System response on load throw off

The analysis of these problems based on the design data (Annexure 8.1), past operating data (Annexure 8.2) and the data collected during the audit is as follows:

8.2 High auxiliary power consumption

8.2.1 New CPP

Data on monthly auxiliary power consumption was collected for last six months (Table 8.2.1).

Table 8.2.1 Auxiliary power consumption variation

Month	*Generation (lac units)*	*Average MW*	*Auxiliary consumption (lac units)*	*% Auxiliary consumption*	*Plant load factor*	*Running hours*
April	132.61	19 73	--	--	73 6	672 25
May	139.96	18.81	--	–	75 2	744 00
June	150.63	20.90	21.95	14.16	83 6	720 00
July	138.91	18.67	21.93	15.8	74 6	744 00
August	128.86	17.73	22 18	17 2	69 3	744 00
September	136 85	19.00	20 52	15 0	76 02	720 00

It can be seen from the table that monthly auxiliary power consumption varies in the range of 14.6 to 17.2%.

8.2.1.1 Measurement

Measurements of individual auxiliary load installed in new CPP was carried out using portable power analyser to estimate the auxiliary power requirement of the plant. Details of the measurements are given in Annexure 8.3. It was found that auxiliary power consumption was 16.6% for a plant load factor of 50%.

The accuracy of the energy meter installed at the load centre 3 (receiving end) was also checked with portable power analyser. It was found that energy meter was showing correct reading.

8.2.1.2 Findings and analysis

Investigation revealed that operationally plant is working at optimum value of auxiliary consumption. Following reasons were identified for higher auxiliary power consumption of the plant.

- Power distribution loss between new CPP and load centre-3 is also added to auxiliary power consumption of the plant. New CPP supplies power to load centre-3, which is 550 m away from plant. Cable details were collected to calculate this distribution loss. It has been worked out that distribution loss is 0.37% of the total auxiliary power consumption. Detailed calculation is given in Annexure 8.4.
- Normally one boiler can provide steam to the TG set; but there are two boilers instead of one for providing steam to TG set. This leads to higher auxiliary power consumption. Power consumed by load connected to one boiler was measured (Annexure 8.3). It was found that load connected to both boilers consumes 5% auxiliary power. Power consumption by single boiler instead of two would have been in the range of 3-3.5%. Therefore, additional auxiliary power to the extent of 1.5% is consumed by loads connected to both the boilers.
- Cooling towers consume additional power to the extent of 0.25%.
- Plant operates at load factor of 60 to 80%. Auxiliary power consumption is high at low PLF. The variation of auxiliary power consumption with respect to plant load factor is shown graphically in figure 8.2.1.2. It can be seen from the figure that the auxiliary power consumption will reduce slightly at full load.
- Auxiliary consumption can be reduced by 0.25% if the CHP operates at its designed capacity (140 tons) instead of present rate of 50 tons/hr.
- Auxiliary consumption will further get reduced by 0.99% if the effect of dumped steam is taken into account by including the equivalent generation loss in the total generation (Table 8.2.1.2a).

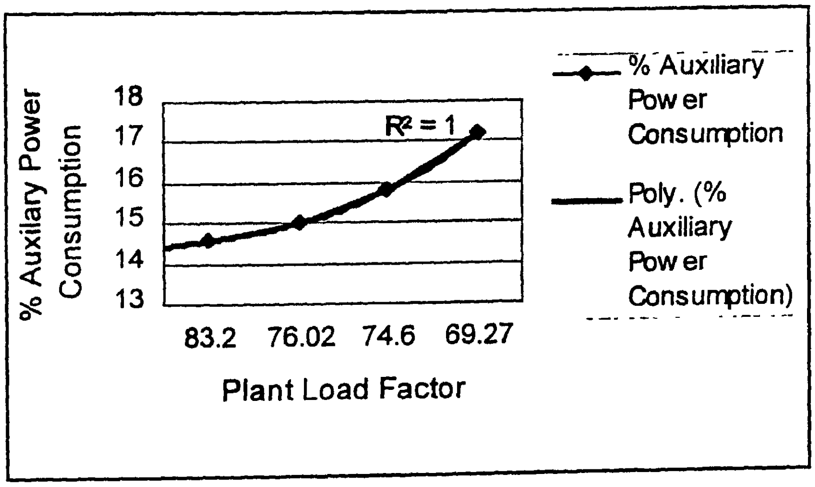

Figure 8.2.1.2 Auxiliary power consumption at different plant load factors

Table 8.2.1.2a Generation and Equivalent dump steam losses (April to September 2001)

Month	Generation lakhs unit	Generation loss due to dump steam lakhs unit	Total units	Auxiliary consumption (lakh units)	% auxiliary consumption		Difference
					without dumping loss	With dumping loss	
April	132.61	7.24	139.80				
May	139.96	7.73	147.69				
June	150.63	7.23	157.86	21.95	14.57	13.90	0.67
July	138.91	12.17	152.08	21.93	15.78	14.51	1.27
August	128.86	8.38	137.24	22.18	17.21	16.16	1.05
September	136.85	9.52	146.37	20.52	14.99	14.01	0.98
Total	827.82	52.27	881.04			Average	0.99
Average generation loss due to dumping				-	5.9%		

Balance sheet of auxiliary power consumption is given in Table 8.2.1.2b.

Table 8.2.1.2b Balance sheet of auxiliary power consumption

Present auxiliary power consumption	-	15.0 %
Higher aux. Consumption due to		
Distribution loss	-	0.37%
2 boilers instead of one	-	1.50%
CT pumps and fans	-	0.25%
CHP	-	0.25%
Partial loading	-	0.25%
Dumping steam effect	-	0.99%
Total	-	**3.61%**
Net auxiliary consumption after considering all the contributing factors	-	11.39%

8.2.2 *Old CPP*

Total auxiliary power consumption as well as the power consumption of individual load in the old CPP plant was measured using portable power analyser. Measurements for total auxiliary power consumption of the plant is given in table 8.2.2.

Table 8.2.2 Auxiliary power consumption

Location/load	Voltage (kV)	Current (Amp)	Power (kW)
2 5 MVA transformer	3 3	54	202
2 MVA transformer	3 3	39	180
CWP-2	3 3	28.66	140.7
Total auxiliary power consumption			522
Load on TG set	3 3	420	2200
Percentage auxiliary load			23.7%

It can be seen from the table that auxiliary power consumption was 23.7% at a plant load factor of 55%.

8.3 High percentage of unburnt combustible in fly ash

Operational records of September 2001 show that fly ash constitutes 70% and bottom ash 30% of the total ash. The unburnt in bottom ash is around 3% which is reasonable. The unburnt carbon in the flyash however varies in the range of 12-18% at different loads as given in table 8.3. The graph of percentage combustibles in the fly ash against the load (Figure 8.3) does not establish any correlation between the two. However, the values are high at both low and high loads.

Table 8.3 Percentage of unburnt carbon in fly ash

	Combustible (%)		Average	Load	Date	Combustible (%)		Average
			load	(%		Boiler		load
Date	Boiler 1	Boiler 2	(MW)	MCR)		1	Boiler 2	(MW)
Sep 01	13 00	11.40	17.70		Sep 16	13.80	13.30	19.75
02	14 20	16.20	16.67		17	13.50	18.40	20.17
03	15.30	15.10	19.29		18	09.50	10.00	19.70
04	13.80	14.90	18.54		19	11.00	09.80	20.54
05	15.80	13.70	18.80		20	10.20	12.50	22.16
06	15 60	14.20	19.95		21	11.80	10.60	21.90
07	15.00	13.80	19.60		22	12.60	11.60	11.50
08	17.00	17.90	20.12		23	12.70	-	09.12
09	13.20	17.60	20.46		24	11.90	14.00	20.45
10	15.00	18.00	15.67		25	12.00	12.50	20.33
11	16.30	17.70	22.80		26	12.70	13.00	20.16
12	16.40	17.00	20.00		27	10.30	13.00	-
13	12.30	15.60	19.00		28	12.20	14.40	21.80
14	1270	17.00	17.90		29	14.40	12.20	-
15	10.80	14.20	19.50		30	10.50	15.20	-

The main reasons for the high unburnt combustibles in the fly ash is due to the following.

1. *High percentage of fines in the coal fed to the boilers*

   The sieve analysis for the crushed coal to be fed to the boiler (as given in the specifications of the boiler) is 100% passing through 10 mm screen and 50% passing through 3.3 mm screen. But this is not so in the case of coal being fed, as is given in the Table 2.2.1. This analysis shows that 67.2% passes through 3.3 mm screen and further 49.7% passes through 1 mm screen. This means that there are too much fines in the coal.

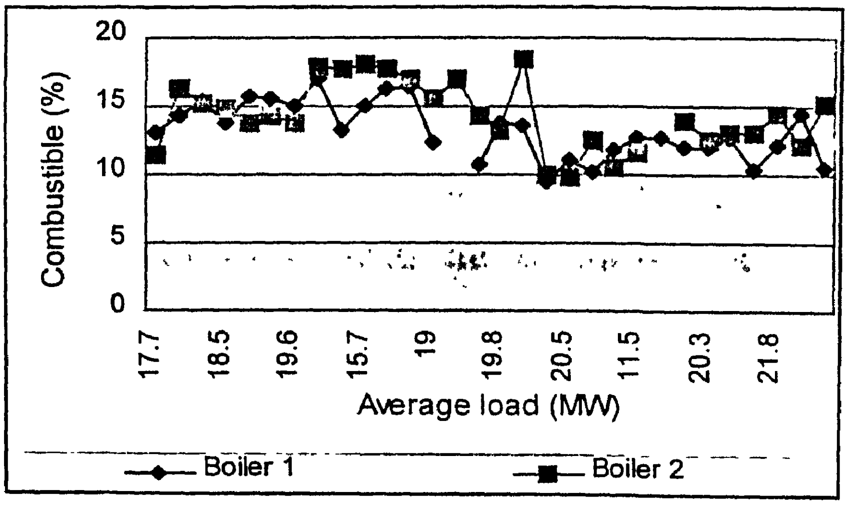

Figure 8.3 Variation of unburnt carbon with load

2. *Overbed coal feeding system*

   In the feeding system provided, the crushed coal is spread over the fluidised bed by spreader stokers. Thus when the coal falls over the bed, the fine particles are not able to penetrate the bed as the flue gas velocity at that point (boiler is designed for fluidising velocity range of 2 to 2.2 m/s) is more than the elutriation velocity of these fine particles. These particles get a chance to burn only in the free board.

3. *Small free-board height in the furnace*

   The free-board in the furnace is as follows:

   First 1000 mm height above the bed is free cross-section; further from 1000 mm to 4500 mm (approx) till the tube of SH-II, it is sloped to half the furnace cross-section. So effective free-board height is only 1000 mm, as in the sloping section, the flue gas velocity goes on increasing to twice its original value which will further escalate elutriation.

The overall residence time for combustion in the free-board may be about 1 sec. However, in literature, it is suggested that a residence time of at least 2 sec is required in the free-board for good combustion.

In view of the above, it is suggested that
(a) As a first priority, the coal feeding should be converted to under-bed feeding system
(b) Alongwith this modification, the boiler designer/supplier may also be asked to look into the possibility of increasing the free-board height. This would be possible only through lowering the level of the distributor plate of the fluidised bed furnace. The air plenum in that case would be at below ground level, and the FD fan ducting would have to be routed suitably.
(c) It would not be practically possible to control the percentage of fines in the crushed coal.

If modification (a) and (b) above are carried out, it may be possible to reduce the unburnt carbon in the fly ash to the level of 4 to 6%.

8.4 High heat rate

The station heat rate is generally calculated as per procedure adopted in section 2.3.1 and the GCV of the coal sampled (Case A). In this case, the GCV recorded was 4680 kCal/kg. The heat rate works out as 3277 kCal/kWh.

However, as per the log book data (table 2.3a), it works out to be 3432 kCal/kWh (taking average GCV of coal as 3900 kCal/kg). Let us call this Case B.

For analysis purpose, let us not take note of difference in GCV. Any correction in Case B will further increase the difference between the heat rates of Case A and Case B.

We notice that only difference in the operating condition of Case A and Case B is that there was no steam dumping in case of A, whereas in Case B, the heat loss due to dumping of the steam if also included. This means that there is loss of heat rate due to dumping of steam and this occurs frequently. The average difference is 273 kCal/kWh (approx. 8.33%) even when we do simple calculations as given in table 2.3b.

Thus, it is obvious that if we want to improve the heat rate, the steam dumping should be avoided, and proper load management of the cement plant (particularly the raw mill) should be taken up on highest priority.

The second factor which contributes significantly to the station heat rate (calculated on the basis of coal consumption) is the loss of combustibles in the fly ash which directly affects the boiler efficiency. This can be improved through suggestion given in section 8.3.

Another factor which can contribute to high heat rate is the make-up water. Its value is 4.3% as given in Annexure 8.2. Acceptable norm is 1% to 1.5%. Excess make up contributes to 3.3 tons of waste steam/ per hour (2.5% of 132 tons). Average loss of heat at 15 kg/cm^2 pressure will be 666 kCal/kg. Thus, the heat loss per kWh comes to 80 kilo calories. It is thus recommended that the boiler may be operated on intermittent blow-down rather than continuous blow-down, to maintain 4 ppm silica in the boiler water (and other chemical regimes as necessary, phosphate – 30 ppm, conductivity 100, pH 9.5 – 10.2). Since the quality of DM water is well maintained and there is no condenser leakage, the intermittent blow-down may not pose any problem.

8.5 Improvement of power factor beyond 0.88

The current required by electrical loads such as induction motors, fluorescent light, induction heating furnace, transformers etc can be considered to be made up of two separate kinds of current -magnetising current and the power producing current. Some load such as incandescent lamps requires only the power producing current.

Power producing current is that current which is converted into useful work such as turning the lathe, pumping water etc. The unit of measurement of power produced is kW.

Magnetising current is that component which is required to produce the flux necessary for the operation of induction devices. Without magnetising current energy cannot flow through the core of the transformer or across air gap of an Induction motors. The unit of measurement of magnetising volt-amperes is (kVAR).

The total current is the current that is read on an ammeter in the circuit. It is generally made up of both magnetising current and power producing current.

The power factor is expressed as the ratio between power producing current in a circuit to the total current in that circuit. Another definition of power factor is the ratio of kW or working power to that of the total kVA or apparent power.

When the power factor of a load is improved, the total current is reduced (presuming that the load is kept constant) as there is a reduction in kVAR required from the source (utility, captive power generating station). Therefore there is a release of system capacity of (cables, transformers etc), reduction in cable losses because of the reduction in total current and finally improvement in the voltage profile as there is a reduction of voltage drop due to the impedance in the circuit.

There are various ways of improving the power factor, the most common method used in the industry are installation of capacitor banks. These banks supply the kVAR requirement of the load locally leading to a reduction in the total power drawn from the source.

8.5.1 *Concept of Power factor improvement as applied to CPP*

Some the power factor improvement benefits in power plants are detailed in following sections.

8.5.2 *Reduction of auxiliary system loading*

Improving the power factor in auxiliary equipments like pumps, fans and compressors etc. helps in releasing some of the system capacity. This release of existing transformers switchgears and feeder capacity may seem unnecessary for a plant operating as it was initially designed to operate. However power plants which are forced to backfit certain large electrical devices to accommodate, for instance environmental requirements in fossil fuelled plants, or modernise some components of the system. In such cases this released capacity may just be necessary to avoid an expansive auxiliary system expansion.

8.5.3 *Reduced system losses*

The principal sources of energy losses in a power plant auxiliary system are the transformers. The load loss component in the transformation losses is proportional to the kVA loading, the reduction in kVA loading of the auxiliary system aids in reducing the losses in transformers.

8.5.4 *Improvement of auxiliary system voltage profiles*

The reduced kVA loading of the auxiliary system results in a reduced voltage drop through the system impedance's, particularly the step down transformers. This results in improved voltage profiles.

8.5.5 *Release in generating capacity*

If a generator is operating to supply the system load requirements at less than its rated power factor, there is unused turbine capability to produce real power which is unavailable because of the need to support the reactive power requirements. This release in unused turbine capacity is clearly dependent on the generator capability curve. This logic however does not apply if the generator in question is operating at or above the rated power factor. Most turbine-generator sets are turbine limited or output is restricted by the main transformer capacity.

It is suggested that the addition of capacitors to power plant which are presently reactive power limited could realise generally an incremental power output gain upto 1% of the generating nameplate rating. This should be at a cost less then that required for an equivalent new installed generating capacity. Further no additional transmission capacity would be required for this "boost" in real power output.

8.5.6 *Power factor improvement of the generator at new power plant at ACC Keymore*

The Generator of the new plant is rated for 31250 kVA at a power factor of 0.8. The generator losses at 0.8 and unity power factors are given in table 1and 2. These are obtained from design documents provided by ABB. It is observed that there is a maximum reduction of (30.5kW) in generator losses for improving the power factors from 0.8 to 1.

Table 8.5.6a Efficiency of the generator at various operating loads at 0.8 and unity power factor.

%load	100	75	50	25
Effy of gen at unity pf	98.78	98.73	98.47	97.42
Effy of gen at 0.8 pf	98.36	98.3	97.99	96.7
Improvements in efficiency	0.42	0.43	0.48	0.72

Table8.5.6b Losses (kW) of the generator at various operating loads at 0.8 and unity power factor

%load	100	75	50	25
Loss (kW) of gen at unity pf	386.1	302.3	242.6	206.8
Loss (kW) of gen at 0.8 pf	416.6	323.4	255.9	213.1
Reduction in losses (kW)	30.5	21.1	13.3	6.3

The plant is presently operating at a power factor of 0.88. The improvements have been carried out by installing capacitor banks at different load terminals. Therefore they have been able to avail the benefits due to improvement in the power factor explained above like reduction in losses, improvement in the voltage profiles, etc.

However, for considering further power factor improvements from 0.88 to unity, the following problems should be studied thoroughly.

8.5.7 Effect on Stability of the system

By changing the operating power factor of the load, the operating point in the capability curve would change. Both the dynamic as well as transient stability performance could be effected. Particularly in the case of the load throw off of 6MW(tripping of the raw mill).

8.5.8 Problems due to reactive compensation in the context of load throw-offs

The improvements of power factor would involve additional reactive compensation. If these are in the form of medium voltage capacitor banks then frequent switching of these banks with the change in the loads could create additional problems like voltage transients. On the other hand if these banks remain in circuit at periods of low load conditions, there could be problem because of over compensation (rise in the voltage levels).

Precise answers to the above problems can only be given by carrying out a very detailed stability study. It is therefore recommended that the plant should carry out a detailed system study before considering further power factor improvement. For this study, the generator supplier (designer) should also be involved.

8.6 Boiler operation at low loads

In the new CPP, two boilers have been provided; each one to contribute 50% of steam to generate full power of 25 MW. The full steam generation capacity of each boiler is 66 ton per hour at 66 kg/cm^2 (g) pressure which can run the TG upto 12.5 MW. However, it is observed that on 17.10.2001, the cement plant had its one raw mill tripped (under maintenance). The load requirement came down to 11 MW in new CPP. Both the boilers were kept in operation at 16 MW equivalent load out of which 11 MW was electrical generation and balance 5 MW equivalent steam was being dumped. Ideally, only one boiler should be operated under such conditions and no steam should be dumped. Alternately, two boilers could be operated stably at 50% of its capacity. This can be possible if the air dampers are air tight and the 50% of the fluidised bed is slumped. Under these conditions, the load can be raised quickly when required. It is suggested that problem connected with the passing of air dampers may be studied during annual maintenance and the damper made air tight. Then slumped bed operation during part loads can be tried and operators' confidence in such modes of operation can be picked up during the course of time. This will help in reducing long duration of operation with dump steam, and thus improving the overall heat rate of the plant.

8.7 Integrated operation of new and old CPPs

For the purpose of providing power supply to the cement plant and meet all other requirements, ACC has two captive plants, viz., old and new. The particulars of these plants are provided in table 8.7.

Table 8.7 Particulars of old and new CPPs

	Old	*New*	*Remarks*
Boiler			
Stocker fired - 2 nos.	Each of 25 tones/hr, 33 kg/ cm^2 steam pressure	--	Installed in 1950
FBC - 2 nos.	25 tons/hr, 33 kg steam pressure	-	Retrofitted in 1991 from Stocker to FBC
AFBC - 3 nos.	--	Each of 66 tons, 66 kg/cm^2 steam pressure	Installed in 1998
Turbines			
3 nos.	6 MW, 8.6 MW and 6 MW	--	ABB 1950, 1956 make
1 no.	--	25 MW	ABB make

7.1 *Configuration*

8.7.1.1 Old plant

All the boilers output is connected to common header, and can feed any of the three turbines.

8.7.1.2 New plant

The three boilers each of 66 tons/66 kg ratings are provided so that two of them provide steam to turbine to generate 25 MW power and the one act as a reserve.

Steam and condensate lines of the new plant has been connected to the old plant as shown in figure 8.7.1.2.

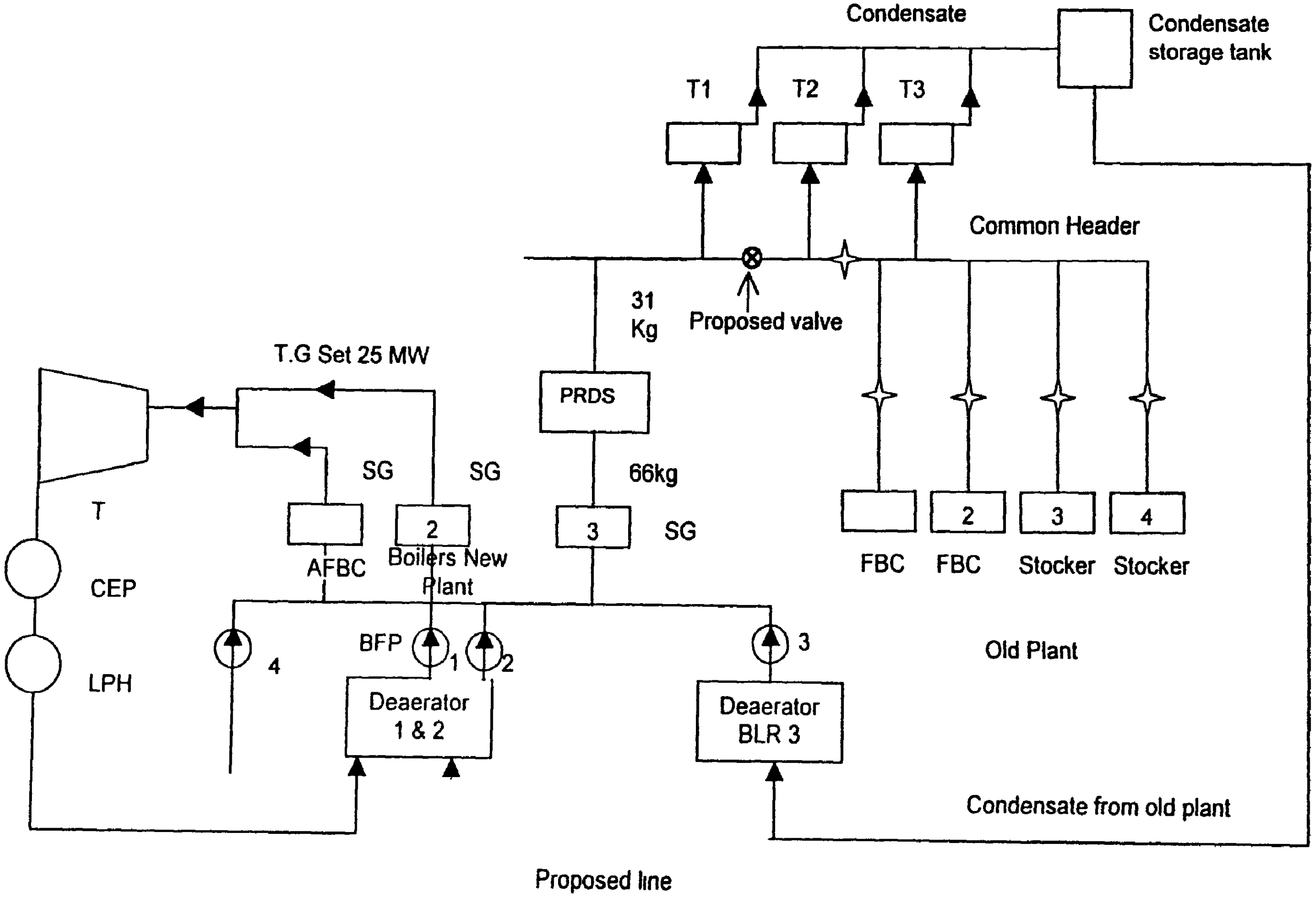

Figure 8.7.1.2 Integrated steam-water circuits of two CPPs

The existing system has the following limitations:

1. Steam from the new plant can be supplied to run turbine T_1 and T_2 of old plant (and not T_3)
2. Condensate from the transfer pump (old) can only be transferred to Deaerator of new plant connected to boiler 3 (and not to deaerator of boiler 1 & 2).

8.7.2 Proposal to improve the existing system

1. One additional valve in the steam header of old plant in between T_1 and T_2 be provided so that main steam from new plant can be fed to any one of the three turbines of the old plant.
2. Deaerator of SG1 and SG2 of new plant be connected to deaerator of SG3 with isolating valve in between so that condensate from the old plant can be transferred to either of the deaerator of the new plant.

With the above modification, spare capacity of the SG1 and SG2 available can be utilised to run 6 MW turbine of old plant. This adds up total generation to 31 MW. Thus, only 7 MW additional power is required. There are two FBC boilers each of 25 ton/31 kg cm^2 pressure making total of 50 ton steam capacity at 31 kg. Plant can generate the balance 7 MW using steam from one of these boilers. The above analysis suggests that it is possible to meet the entire demand of 38 MW with SG3 and both the stoker-fired boilers still in reserve.

8.7.3 Renovation/modernisation/retrofitting of old plant - Future requirements

Old plant is about 50 years old. Life assessment studies indicate 10 years life still left out. For future requirement beyond 10 years, it is suggested to add one more boiler of 66 ton/66 kg/cm^2 capacity and two turbines of 15 MW capacity. This will help to have spare capacity of 15 MW (both boiler and turbine) which will serve as reserve or spare power for any future load. The old power plant may thus be shut down and sold off. This will improve the overall heat rate. However, a separate techno-economic study may be conducted for the additional cost (after selling the old plant) to be incurred and the fuel saving through the improvement of heat rate.

8.8 Operation management

1. The 25 MW new CPP operates in conjunction with old CPP having three turbines of 6, 8.6 and 6 MW.
2. Load excursions
   The 25 MW plant has been provided with steam dumping device to help raise steam parameter during start up and dump the steam at the time of load throw off and also for readiness to take sudden load of 6 MW. The operation records (table 8.2.1.2a) indicate that there has been steam dumping of the order of 5.9% on account of tripping of raw mills and other outages of the cement plant. The average number of starts and stops per month during the last six months were 198, i.e. 6 per day. These outages not only result in lower plant efficiency reflected by way of high heat rate but adds to complexities of operation like temperature excursion and high drum level with a chance for carry over. Since the plant operates in radial mode (i.e. isolated from grid) and is subjected to severe operational duty, it is necessary to develop "Operation and Performance Monitoring Guidelines".

The sustained high level of performance and efficiency in a power plant depends on close monitoring of plant operation and analysis of deviations in the parameters from the set/designed or rated values. This calls for developing systems, imparting right type of training, and providing manuals, operating instructions and schematics to engineers for right analysis, action planning and execution of various jobs. Keeping in view the need to maintain high plant efficiency and availability, the following suggestions are made.

1. Revision of manuals, operating instruction, and drawings as per present operating conditions. This is important for operating the plant reliably and efficiently. These documents should include operating principle, exploded view of equipments, start up/shut down instructions, equipment details, specifications, optimum operating values, excess margins, and do's and don'ts;
2. Printed single line drawing for all the main and sub-systems;
3. Technical diary to include specifications and technical details of each and every equipment and system;
4. A separate engineer may be designated as Operations and Efficiency Engineer to work as a nodal point for all operational and efficiency works.

5. A separate coal engineer supervisor to supervise operation of Coal Handling Plant and its maintenance, ensure proper sampling, analysis of coal and ash, and carry out modification and follow up of approved plans.
6. Develop maintenance philosophy and maintenance planning systems to ensure timely availability of spares, and execution of planned and emergency maintenance jobs.
7. Water chemistry is a very important area. All the shift charge engineers must familiarise with the various parameters, the blow down requirement, and the phosphate and hydrazine dosing instructions. A record of such instructions should be kept in the control room and countersigned by shift charge engineers.

8.9 System response on load throw off

The incidence of load throw off is very high. The number of such incidences are as high as ten in a day. The extent of load throw-off varies from 5 MW to 12 MW. Since the power system is not connected to the grid, these load changes lead to fluctuations in voltage and frequency.

We have analysed one load throw off of 5 MW magnitude on load on 20th October. The print out was taken and is enclosed as Annexure 8.4. The measured values were as follows.

		Before	After
Load drop	-	21.41 MW	16.45 MW
Condenser vacuum	-	-0.91	-0.90
Frequency	-	50.14	50.26
Turbine inlet pressure kg/cm^2	-	61.83	64.35

It is observed that all the fluctuations are well within the safe limits. Steam dumping device worked well this time. But this may not be the case always. Sometimes the voltage and frequency fluctuations may be very high and may lead to tripping (one incident of tripping was also noticed during the week our audit was being conducted; at that instance, unit could be brought back in about 7 hours).

It is thus suggested that the response of the governing system and the voltage regulator under different load throw-offs should be analysed, and the power station is operated within the safe limits of load fluctuations. This will improve the reliability of the operation of the power plant and thus the productivity of the cement plant.

Annexure 2.1

Boiler efficiency calculations

Parameters	Boiler#1	Boiler#1
Oxygen in flue gas (%)	3 25	4.0
Carbon monoxide in flue gas (PPM)	250	450
Carbon dioxide in flue gas (%)	15 7	14.93
Excess air (%)	17	24
Flue gas temperature (°C)	154	123.5
Theoretical air requirement (kg/kg of fuel)	7.13	6.52
N2 in flue gas (kg/kg of fuel)	81.05	81.1
Dry flue gas quantity (kg/kg of fuel)	7.68	8.05
Nitrogen quantity in flue gas (kg/kg of fuel)	5.69	5.99
Inlet air quantity (kg/kg of fuel)	7.40	7.79
Heat released by burning 1 kg of carbon (kcal/kg)	8084	8084
Mean specific heat of flue gas (KJ/kg C)		
	1.0100	1.0058
Moisture calculation		
Air		
DBT (°C)		
WBT (°C)	30	30
Humidity ratio (kg/kg of dry air)	25.3	25.3
Moisture content of air (kg/kg of air)	0.0184	0.0184
GCV of coal(as fired)	0.0181	0.0181
Fuel	4648	4648
Moisture in fuel (kg per kg of fuel)		
Hydrogen in fuel (kg per kg of fuel)	0.085	0.085
Total moisture in flue gas (kg per kg of fuel)	0.04026	0.04026
Specific heat of moisture (Kcal/kg-°C)	0.578	0.578
	0.4498	0.4498
Losses		
Dry flue gas loss	4.93	3.89
Heat loss due to CO in flue gas	0.001	0.002
Heat loss due to moisture in air	0.16	0.13
Heat loss due to moisture and hydrogen in the fuel	6.10	5.97
Heat loss due to unburnt in bottom ash		
Heat loss due to unburnt in fly ash	0.64	0.64
Sensible heat loss in bottom ash	6.15	6.15
Sensible heat loss in fly ash	0.25	0.25
Radiation heat loss	0.16	0.12
	0.39	0.39
Total Losses		
Thermal Efficiency	**18.79**	**17.55**
	81.21	**82.45**

Annexure 2.2

Savings due to plugging the air leakage in Boiler#1

Existing case

Measured temperature in boiler#1 = 135°C

Measured air infiltration = 22 %

Actual temperature in case of no infiltration = 153 °C

Existing efficiency = 81.21 %

Existing monthly coal consumption = 6000 T

Improved conditions (plugging the infiltration)

Anticipated improved efficiency = 82.10 %

Reduction in coal consumption = 49 T/ month
(assuming that 75% of the total time it operates on full load)

Annual coal savings = 392 T/annum
(Total operating months per annum=2/3)

Monetary savings per annum = Rs. 4.7 lacs
(@Rs. 1200/T of coal)

Annexure 2.3

Boiler efficiency

Parameters	Boiler#2	Boiler#3	Boiler#4
Oxygen in flue gas (%)	6.05	8.44	10.28
Carbon monoxide in flue gas (PPM)	1837	38	38
Carbon dioxide in flue gas (%)	12.71	10.75	9.17
Excess air (%)	38	64	91
Flue gas temperature (°C)	251	288	251
Theoretical air requirement (kg/kg of fuel)	6.54	6.39	6.39
N2 in flue gas (kg/kg of fuel)	81.23	80.81	80.56
Dry flue gas quantity (kg/kg of fuel)	9.35	10.82	12.61
Nitrogen quantity in flue gas (kg/kg of fuel)	7.02	8.15	9.52
Inlet air quantity (kg/kg of fuel)	9.14	10.60	12.39
Heat released by burning 1 kg of carbon (kcal/kg)	8084	8084	8084
Mean specific heat of flue gas (KJ/kg C)			
	1.0302	1.0371	1.0308
Moisture calculation			
Air			
DBT (°C)			
WBT (°C)	30	30	30
Humidity ratio (kg/kg of dry air)	25.3	25.3	25.3
Moisture content of air (kg/kg of air)	0.0184	0.0184	0.0184
GCV of coal(as fired)	0.0181	0.0181	0.0181
Fuel	4685	4538	4538
Moisture in fuel (kg per kg of fuel)			
Hydrogen in fuel (kg per kg of fuel)	0.093	0.085	0.085
Total moisture in flue gas (kg per kg of fuel)	0.040815	0.039345	0.039345
Specific heat of moisture (Kcal/kg-°C)	0.623	0.628	0.660
	0.4498	0.4498	0.4498
Losses			
Dry flue gas loss			
Heat loss due to CO in flue gas	10.87	15.22	15.10
Heat loss due to moisture in air	0.01	0.000	0.000
Heat loss due to moisture and hydrogen in the fuel	0.35	0.49	0.49
Heat loss due to unburnt in bottom ash	6.66	6.72	6.56
Heat loss due to unburnt in fly ash			
Sensible heat loss in bottom ash	0.06	8.86	8.86
Sensible heat loss in fly ash	6.39	0.00	0.00
Radiation heat loss	0.12	0.26	0.26
	0.22	0.00	0.00
Total Losses	0.50	0.50	0.50
Thermal Efficiency			
	24.69	**31.55**	**31.27**
	75.31	**68.45**	**68.73**

Annexure 2.4

Savings due to plugging the air leakage in Boilers

	Boiler#2	Boiler#3	Boiler#4
Existing case			
Measured temperature in boiler before ID fan (°C)	202	130	130
Measured excess air (%)			
after boiler	38	64	38
before ID fan	82	312	91
Existing efficiency (%)	75.31	68.45	68.73
Existing monthly coal consumption (T/hr)	5.71	1.90	1.75
Improved conditions (plugging the infiltration)			
Anticipated temperature (°C)			
Desired conditions	251	288	251
excess air range (%)			
flue gas temperature (°C)	5	8	8
Anticipated improved efficiency (%)	150	150	150
Reduction in coal consumption (T/hr)	81.4	77.61	77.84
Annual coal savings (T/year)	0.43	0.22	0.20
(Assuming 4000 working hrs per year)	1708	897	818
Monetary savings per annum (Rs. Lakh)			
(@Rs. 1200/T of coal)	20.51	10.77	9.82

Annexure 4.1

Estimation of total compressed air leakage in ash handling line 2 and 3 (Using compressor #2)

Specific power consumption (based on actual FAD) of compressor #2 = 1.36 kW/m^3/min

Table A4.4 Total compressed air leakage in ash handling line 2 and 3

Sl. no	Parameter	*Level for line #2*	*Level for line #3*
1.	Initial pressure in the gauge on receiver, kg/cm^2 (g)	3.2	3.7
2.	Final pressure in the gauge on receiver, kg/cm^2 (g)	3.8	3.75
3.	Volume of air receiver, m^3	15.0	15.0
4.	Free air delivery of compressor (designed), m^3/min	10.2	10.2
5.	Average time taken to fall in air pressure in air receiver, minutes	10.59 (3.8 - 3.2 kg/cm^2,g)	14.10 (3.75 - 3.7 kg/cm^2,g)
6.	Average time to built up air pressure in air receiver, minutes	1.11 (3.2 - 3.8 kg/cm^2,g)	1.02 (3.7 - 3.75 kg/cm^2,g)
7.			
8.	Compressed air leakage, %	9.5	6.7
9.	Compressed air leakage (based on actual FAD), m^3/min (or cfm)	0.67 (~ 23.8)	0.48 (~16.9)
10.	Loss due to leakage, kW	0.92	0.65
Arresting the leakage and allowing only 5% leakage in the AH 2 and 3 can save about 53% and 74% of this loss.			
	Electrical savings per annum (@ 5600 working hr per annum), kWh	2430	940
	Monetary savings per annum (@ Rs 2.00/kWh), Rs.	4900	1900

Annexure 5.1

Replacement of aluminium blades of cooling towers fan with hollow FRP blades

Table A5.1 Savings due to replacement of aluminium blades of cooling tower

Sl. no.	Parameter	*CT #1*	*CT #2*
1	Number of fan in cooling tower	4	3
2	Installed capacity of motor for the fan in cooling tower (kW)	30.00	30.00
3	Avg.operating load of the fan motor (%)	93.00	91.00
	Avg.operating load on the fan motor (kW)	27.90	27.30
4	Anticipated savings by using hollow FRP blades (%)	15	15
5	Actual Savings (kW)	4.185	4.095
6	Assumed working hours (hrs) per annum	7008	7008
7	Savings (kWh) per annum	29328	28698
8	Monetary savings per annum, Rs. (@ Rs. 2.00/kWh)	58657	57396
9	Estimated investment (Rs)	45000	45000
10	Simple pay back period (months)	9.2	9.4

Annexure 5.2

Cooling towers installation with CT-ENSAVER+ (thermostatic controller)

By installing the CT-ENSAVER+ (thermostatic controller) for fan motor circuit in cooling towers, the approximate savings and reduction of operating hours per annum per fan is different.

Table A5.2 Savings due to installation of CT-ENSAVER+ (thermostatic controller)

Sl. No	Parameter	*CT #1*	*CT #2*
1	Number of fan in cooling tower	4	3
2	Installed capacity of motor for the fan in cooling tower (kW)	30.00	30.00
3	Avg.operating load of the fan motor (%)	93.00	91.00
	Avg.operating load on the fan motor (kW)	27.90	27.30
	Assumed working hours (hrs) per annum *	7008	7008
4	Assumed working hours reduction	2150	2150
5	Savings (kWh) per annum	59996	58706
6	Annual monetary savings, Rs. (@ Rs. 2.00/kWh)	119992	117412
7	Investment (Rs)	35000	35000
8	Simple pay back period (months)	4	4

- Assuming, the CT#1 and #2 runs for 80% of total operating time every day.

Annexure 8.1

Design details

Table A8.1.1 New CPP boiler design data

1	Boiler: ABB make		
	Evaporation at 100%% MCR	-	66 tons per hour (tph)
	SH outlet temperature	-	485°C
	SH outlet pressure	-	66 kg/cm²
	Steam temperature control range	-	50 - 100% MCR
	Design pressure	-	78 kg/ cm²
	Feed water temperature after ECO	-	190°C
	Feed water temperature after HP heater	-	140°C
2.	Turbo generator		
	Condensing type, 3 steam extraction (LP, HP, deaerator), 60 8 Be, 480°C, rpm - 5700, outlet pressure 0.104 bar		
	Generator		
	31.25 MVA, 0 8 PF, 11 kV, 1 640 Amps		
3.	Electrical		
	Station service transformers		
	11 kV/433 volts, 2 MVA	-	3
	Station auxiliary transformer		
	11 kV/6 6 V, 2 MVA	-	2
	Generator transformer		
	11 kV/6.6 kV, 31.5 MVA		

Table A8.1.2 coal proximate analysis

		Range of design
Moisture	13%	10 - 20%
Ash	30%	30 - 45%
V M	25.6%	20 - 26%
Fixed carbon	31.4%	26 - 36%

Ultimate analysis as fired basis

Moisture - 13
Ash - 30
C - 42.58
H_2 - 3.08
Sulphur - 0.6
Nitrogen - 0.86
Oxygen - 6.94
Calorific value - 4000

Initial deformation

Temperature	-	1050°C
Hemispherical temperature	-	>1400°C
Fluid point	-	>1500°C

Coal size

100% through 10 mm mesh
50% through 3.2 mm mesh

Old Plant design details

1. Boiler

Steinmuller Stocker make - 2 nos.

Evaporation

MCR	-	45000 lbs per hr
NCR	-	37000 lbs per hr

FBC (retrofitted)

MCR - 25 tons, 31 kg
445±5 degree

Turbine

3 turbines, Stall make of 6, 8.5, 6 MW capacity

2. Coal handling plant

New Captive Power Plant

Crusher	-	185 tons/hr
Coal size (input crusher)	-	300 mm

Output:

-25 mm	-	100%
-10mm	-	70%
-1 mm	-	20%

Belt capacity after crusher	-	140 tons/hr
Belt capacity at intake	-	185 tons/hr
Belt capacity for return of over size coal	-	45 tons/hr
Hopper capacity	-	500 tons for each boiler (equivalent to 48 hrs of coal requirement at full load)

Steam dump device

Spray valve has three limits

Limit 1	-	1.26 tons/hr
Limit 2	-	2 tons/hr
Limit 3	-	2.34 tons/hr

Annexure 8.2

Past operating data

Table A8.2.1 Water analysis

	Drum water			*Feed*	*Main steam*	
	K1	*K2*	*DM water*	*water*	*Actual*	*Designed*
PH	9.90	9.85	6.88	8.90	9.00	9 - 9.5
Conductivity	68.00	46.00	0.12	0.80	0.16/0.12	<0.20
Silica	0.74	0.60	0.01	0.018	0.016	<0.02
Phosphate	5.20	4.60	--	--	--	--
TDS	44.00	30.00	--	--	--	--
N_2H_4				<0.12	--	--
O_2				<0.007	--	--
Copper	--	--	--	--	--	0.003
Na	--	--	--	--	--	<0.010
Blowdown	--	--	--	--	--	Continuous

Table A8.2.2 DM water consumption

Month	m^3	*Consumption tons/day*
April	3734	125
May	2546	82
June	3845	128
July	4635	150
August	--	--
September	2545	84
Average consumption per day		114 tons
% of rated capacity of boiler		$\frac{114}{2640}$ = 4.3%

Table A8.2.3 Generation cost of NCPP

Month	*PS/kWh*	*Coal cost/MT*
May	131.44	1283.31
June	132.84	1160.26
July	133.69	1181.32
August	153.86	1248.42
September	128.61	1238.39

Coal is received from difference sources. Average coal cost has been worked out on month to month basis.

Annexure 8.3

Auxiliary power consumption

Area	*Equipment*	*Power (kW)*	*Running hours*	*Energy (kWh)*
Boiler-1				
	FDF Motor- A	93	24	2232
	FDF Motor - B	84	24	2016
	IDF Motor - A	24.9	24	597.6
	IDF Motor - B	50	24	1200
	SAF Motor - B	18.9	24	453.6
	Coal Spreader - A	0.66	24	15.84
	Coal Spreader - B	0.43	24	10.32
	Coal Spreader - C	0.6	24	14.4
	Coal Spreader - D	0.51	24	12.24
	Coal Feeder - A	0.15	24	3.6
	Coal Feeder - B	0.12	24	2.88
	Coal Feeder - C	0.12	24	2.88
	Coal Feeder - D	0.45	24	10.8
	BVDC - 1MCC	1.71	24	41.04
	Inst air Compressor	83.5	24	2004
	Sub total	359.05		8617.2
For 2 boilers	Sub total	718.1		17234.4
Combine load of B#1&B#2				
	Lighting Load	2.4	24	57.6
	SW ACDB	0.8	24	19.2
	Air Handling Unit	5.06	24	121.44
	ESP-2 MCC	3.45	24	82.8
	ESP -1 MCC	40.2	24	964.8
	ACW -1	38.7	24	928.8
	AC MCC	23.4	24	561.6
	Ash Handling MCC	77.4	24	1857.6
	HP Dosing Pump	0.18	24	4.32
	Misc MCC	34.2	24	820.8
	Cooling Tower	83.4	24	2001.6
	UPS	0.1	24	2.4
	Water Treatment MCC	10.5	16	168
	ETP MCC	14.4	24	345.6
	TG MCC	64.5	24	1548

Area	*Equipment*	*Power (kW)*	*Running hours*	*Energy (kWh)*
	Ventilation MCC	45.8	24	1099.2
	BFP - 1	255	24	6120
	BFP - 2	249	24	5976
	CWP -3	173	24	4152
	CWP - 2	168	24	4032
Coal Handling Plant				
	Crusher	102	10	1020
	BCN 1 TO 5	38.5	10	385
	BCN 6,7 &8	37.9	10	379
	Sub total	1467.89		32647.76

Total auxiliary power consumption = 49882.16 kWh

Total generation = 12500 x 24 = 300000 kWh

% auxiliary power consumption = 49882.16/300000

= 16.62 %

Annexure 8.4

Distribution loss between New CPP and Load Centre-3

Cable : XLPE unarmoured single core cable

Cable type : A2XCY-11 kV

No. of cores and cross-sectional area of conductor : 1 x 630 sq. mm

Approx. AC resistance : 0.0606 ohms/km

No. of cables : 3 x 3 = 9

Length : 0.55 km

Current in each cable : 533 amps

Distribution loss in each cable : $I^2R = (533)^2 \times 0.0606 \times 0.55/1000$

$= 9.46$ kW

Distribution loss in 9 cables : $9 \times 9.46 = 85$ kW

Distribution loss % of total auxiliary power consumption : $(85/23000) \times 100 = 0.37$